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Development and Demonstration of an Advanced Supermarket Refrigeration/HVAC System

Prepared by:

Foster-Miller, Inc.
350 Second Avenue
Waltham, MA 02451

Principal Investigator:

David H. Walker

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Oak Ridge National Laboratory
P.O. Box 2008
Oak Ridge, TN 37831-6192

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CONTENTS

Section **Page**

EXECUTIVE SUMMARY ES-1

1. INTRODUCTION 1

2. DESCRIPTION OF SUPERMARKET REFRIGERATION SYSTEMS 4

2.1 Refrigerated Display Cases 4
2.2 Walk-In Storage Coolers 7
2.3 Compressor Systems 7
2.3.1 Stand-Alone Refrigeration Systems 8
2.3.2 Multiplex Refrigeration Systems 8
2.4 Compressors 15
2.5 Condensers 16

3. ADVANCED REFRIGERATION SYSTEMS 19

3.1 Low-Charge Multiplex Refrigeration 19
3.2 Distributed Refrigeration 20
3.3 Secondary Loop Refrigeration 24
3.3.1 Secondary Fluids for Supermarket Refrigeration 27
3.4 Advanced Self-Contained Systems 30

4. SUPERMARKET HVAC 33

4.1 HVAC Load Characteristics 33
4.2 HVAC System Configurations 34
4.3 Airflow Path Configurations 36
4.4 HVAC System Equipment 36
4.5 Desiccant Dehumidification 36
4.6 Water-Source Heat Pumps 41

5. ANALYSIS OF SUPERMARKET REFRIGERATION AND HVAC 46

5.1 Multiplex Refrigeration 46
5.1 Analysis Variations for Distributed and Secondary Loop Refrigeration 53
5.2.1 Modeling of the Distributed Refrigeration System 53

Section **Page**

5.2.2 Modeling of Secondary Loop Refrigeration 54
5.3 Heat Reclaim for Multiplex Refrigeration 56
5.4 Supermarket HVAC Analysis 59
5.4.1 HVAC Model Description 59
5.4.2 Analysis of Conventional Air Conditioning Systems 63
5.4.3 Water-Source Heat Pump 65

6. DESCRIPTION OF THE MODELED SUPERMARKET 68

6.1 Refrigeration Description 68
6.2 HVAC Description for the Modeled Supermarket 74
6.2.1 Space Cooling Load 74
6.2.2 Space Heating Load 77
6.3 Utility Rates 79

7. ANALYSIS RESULTS 80

7.1 Refrigeration Analysis Results 80
7.1.1 Impact of Heat Rejection on Energy Consumption 83
7.1.2 Analysis of Refrigeration Heat Reclaim for Space Heating 86
7.2 Evaluation of Environmental Impact of Low Charge Refrigeration 90
7.3 Analysis Results for HVAC 91
7.4 Analysis of Integrated Operation of Refrigeration and HVAC 93
7.5 Payback Analysis for Advanced Systems 96

8. CONCLUSIONS AND RECOMMENDATIONS 99

ILLUSTRATIONS

Figure **Page**

2-1. Layout of a typical supermarket ... 5
2-2. Display case types employed in supermarkets ... 6
2-3. Elements of a stand-alone refrigeration system ... 9
2-4. Diagram of a multiplex refrigeration system ... 10
2-5. Mechanical subcooling of low temperature refrigerant ... 13
2-6. Schematic of hot gas defrost ... 14
2-7. Refrigeration heat reclaim schematic ... 15
3-1. Piping diagram for the low-charge multiplex system ... 20
3-2. Description of the distributed refrigeration system ... 21
3-3. Supermarket layout using a distributed refrigeration system ... 22
3-4. Rejection loop pipe layout for the distributed refrigeration system ... 24
3-5. Elements of the secondary loop refrigeration system ... 25
3-6. Pump power comparison of heat fluids relative to Tyfoxit 1.21 (3-5) ... 30
3-7. Relation between capacity and power used for modeling unloading scroll compressors ... 31
4-1. HVAC load characteristics for supermarkets and other retail buildings (4-1) ... 34
4-2. Elements of a supermarket air handler ... 35
4-3. Rooftop unit for HVAC ... 37
4-4. Single-path HVAC ... 38
4-5. Dual-path HVAC ... 38
4-6. Elements of a desiccant dehumidification system ... 39
4-7. Detailed diagram of the water-source heat pump ... 42
4-8. Integration of supermarket refrigeration and HVAC systems ... 44
4-9. Detailed diagram of water-source heat pump ... 45
5-1. Supermarket refrigeration analysis model ... 47
5-2. Cycle analysis for heat reclaim ... 57
5-3. HVAC model flow chart ... 60
7-1. Annual energy consumption for low-charge supermarket refrigeration systems ... 80
7-2. Analysis of heat reclaim for space heating, multiplex refrigeration with air-cooled condensing, Worcester, MA location ... 88
7-3. Analysis of heat reclaim for space heating, multiplex refrigeration with air-cooled condensing, Los Angeles, CA location ... 89
7-4. Comparison of supermarket HVAC systems ... 94
7-5. Operating cost of supermarket refrigeration and HVAC ... 96

TABLES

Table **Page**

2-1. Typical refrigeration and electric requirements for supermarket display cases 7
4-1. Comparison of first and operating costs for conventional and desiccant HVAC systems (4-5) 40
5-1. Refrigeration system characteristics needed for analysis model input 48
5-2. Heat rejection system specifications for refrigeration modeling 50
6-1. Description of modeled cases and coolers (low temperature) 69
6-2. Multiplex refrigeration system configuration for the modeled supermarket 70
6-3. Distributed refrigeration system configuration for the modeled supermarket 70
6-4. Secondary loop refrigeration system configuration for the modeled supermarket 71
6-5. Heat rejection for modeled refrigeration systems 71
6-6. Parameters used for refrigeration system analysis 72
6-7. System parameters used for the analysis of secondary loop refrigeration 73
6-8. Locations chosen for supermarket refrigeration analysis 74
6-9. Design ambient conditions for the supermarket air conditioning load calculation 77
6-10. Sensible cooling load analysis for supermarket air conditioning 77
6-11. Sensible load analysis for supermarket air conditioning 77
6-12. Analysis of latent loads for supermarket air conditioning 77
6-13. Air conditioning unit sizing for supermarkets 78
6-14. Space heating design ambient temperatures and loads for modeled sites 79
6-15. Utility rates used for the supermarket refrigeration and HVAC analysis 79
7-1. Annual energy consumption (kWh) for low-charge refrigeration systems for selected locations 81
7-2. Annual water consumption for heat rejection distributed refrigeration with an evaporative fluid cooler 82
7-3. Energy savings achieved by low-charge refrigeration systems 82
7-4. Breakdown of refrigeration annual energy consumption (kWh) 83
7-5. Impact of heat rejection on multiplex refrigeration 84
7-6. Impact of heat rejection on distributed refrigeration 84
7-7. Impact of heat rejection on secondary loop refrigeration 84
7-8. Estimated annual energy consumption for standard and low-charge multiplex refrigeration 85
7-9. Annual energy consumption comparison between multiplex with air-cooled condensing and distributed with dry heat rejection 85
7-10. Annual energy consumption comparison between multiplex with evaporative condensing and distributed with evaporative heat rejection 86

Table **Page**

7-11. Annual energy consumption comparison between multiplex and distributed refrigeration with water-cooled condensing and evaporative heat rejection 87
7-12. Heat reclaim performance multiplex refrigeration with air-cooled condensing Worcester, MA location 88
7-13. Heat reclaim performance multiplex refrigeration with air-cooled condensing Los Angeles, CA location 89
7-14. Total Equivalent Warming Impact (TEWI) for supermarket refrigeration 91
7-15. Estimated operating cost savings for reduced refrigerant leakage 92
7-16. Annual energy consumption for conventional supermarket HVAC 92
7-17. Annual energy consumption for water-source heat pump HVAC 92
7-18. Annual operating costs for supermarket HVAC 94
7-19. HVAC operating cost savings achieved by water-source heat pumps 95
7-20. Estimated annual operating cost for supermarket refrigeration and HVAC 95
7-21. Annual cost savings achieved by distributed refrigeration and water-source heat pumps versus multiplex refrigeration and conventional HVAC 96
7-22. Estimated installed cost premiums for low-charge supermarket refrigeration systems ... 97
7-23. Estimated installed cost premiums for water-source heat pumps 97
7-24. Estimated energy savings and payback for low-charge supermarket refrigeration (versus multiplex with air-cooled condensing) 98
7-25. Estimated payback for distributed refrigeration and water-source heat pumps 98

EXECUTIVE SUMMARY

Supermarkets are the largest users of energy in the commercial sector with a typical supermarket consuming on the order of 2 million kWh annually. One of the largest uses of energy in supermarkets is for refrigeration, which is as much as half of the store's total.

The large majority of U.S. supermarket refrigeration systems employ direct expansion air-refrigerant coils as the evaporators in display cases and coolers. Compressors and condensers are kept in a remote machine room located in the back or on the roof of the store. Piping is provided to supply and return refrigerant to the case fixtures. As a result of using this layout, the amount of refrigerant needed to charge a supermarket refrigeration system is very large. A typical store will require 3,000 to 5,000 lb of refrigerant. The large amount of piping and pipe joints used in supermarket refrigeration also lead to significant leakage, which can amount to a loss of up to 30 percent of the total charge annually.

With increased concern about the impact of refrigerant leakage on global warming, new supermarket system configurations requiring significantly less refrigerant charge are now being considered. Advanced systems of this type include:

- Secondary loop – A secondary refrigerant loop is run between the display cases and a central chiller system. The secondary fluid is refrigerated at the chiller and is then circulated through coils in the display cases where it is used to chill the air in the case.

- Distributed – Multiple compressors are located in cabinets placed on or near the sales floor. The cabinets are close-coupled to the display cases and heat rejection from the cabinets can be done through the use of a glycol loop that connects the cabinets to a fluid cooler, or with direct outdoor air-cooled condensers.

- Advance self-contained - Each display case is equipped with a water-cooled condensing unit. A fluid loop is connected at all condensing units and is used for heat rejection.

- Low-charge multiplex - The multiplex refrigeration system is equipped with control piping and valves to allow operation at close to critical charge, greatly reducing the amount of refrigerant needed.

While these systems use less refrigerant, energy use varies and can be much more than is now seen with centralized multiplex racks. The centralized refrigeration systems are mature in their technology and have been optimized in many ways, in terms of first cost and installation, reliability and maintenance, and energy use.

For an advanced refrigeration system to replace centralized multiplex racks such systems must offer an incentive for their use, such as reduced first cost, good return on investment for first cost, or lower operating and energy costs. Many of the features now used to reduce energy consumption of the centralized systems could be employed with the advanced systems to increase their energy efficiency. Examples include reduced head pressure and mechanical subcooling. The advanced refrigeration systems also have inherent characteristics, which could lead to reduced energy consumption if these features can be utilized as much as possible. An example is that both the secondary loop and distributed refrigeration systems employ significantly shorter refrigerant suction lines (close-coupling), which mean that pressure drop and heat gain is much less than is seen with presently installed central refrigeration systems. Higher suction pressure and lower return gas temperature can translate into lower compressor energy consumption. Scroll compressors used in the distributed refrigeration system (for reduced noise levels) can operate at a lower condensing temperature than reciprocating compressors, because scroll compressors have no suction valves. This feature could allow the distributed refrigeration system to operate at a much lower head pressure (i.e., floating head pressure). Similarly, the charge control method used with the low-charge multiplex system allows the use of lower minimum condensing temperatures with the multiplex system, which lowers energy use during winter operation.

HVAC also represents a large portion of the energy use of a supermarket, on the order of 10 to 20 percent of the store total, depending upon geographic location. The refrigerated fixtures installed in a supermarket have a major impact on the store HVAC.

A possible way to utilize refrigeration reject heat for space heating is through water-source heat pumps, when a glycol/water loop is used for refrigeration heat rejection. In this way the refrigeration reject heat is recovered to provide space heating. This method offers several advantages, which are that a much larger portion of the reject heat can be reclaimed, and the condensing temperature and head pressure of the refrigeration system does not have to be elevated for the heat pumps to use the reject heat. Refrigeration energy savings achieved by low head pressure operation can be realized along with the energy benefits seen through heat reclaim.

An investigation of low charge refrigeration and integrated refrigeration and HVAC was conducted. A model supermarket was formulated and energy consumption estimates were made for present multiplex refrigeration with air-cooled condensing and mechanical subcooling and the advanced, low-charge systems. A similar analysis was performed for the store HVAC where conventional rooftop units, refrigeration heat reclaim, and water-source heat pumps were examined and compared. Four locations with greatly varying ambient conditions were chosen as modeled sites for these analyses.

Figure ES-1 and Table ES-1 give the results for the comparison of refrigeration systems. Results from the analysis showed that the largest energy savings were achieved by the distributed and secondary loop refrigeration systems. The distributed system produced energy savings, ranging from 10.2 to 16.2 percent of multiplex consumption. Secondary loop refrigeration showed reductions in energy of 9.2 to 16.4 percent for the locations investigated. The secondary loop system had higher energy savings than the distributed system for the Memphis and Los

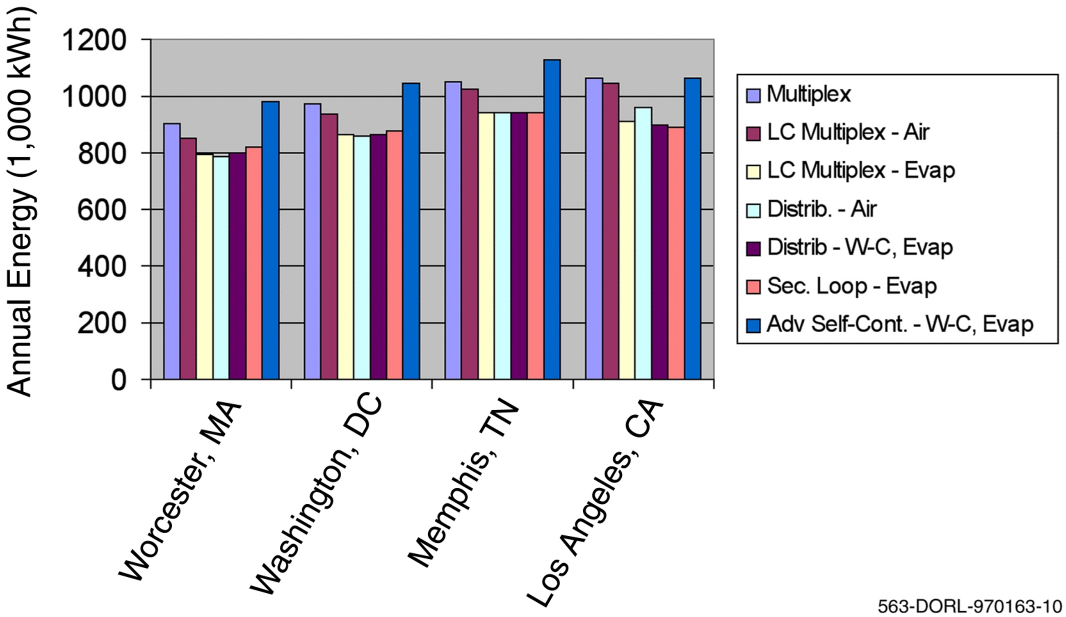

Figure ES-1. Annual energy consumption for low-charge supermarket refrigeration systems

Table ES-1. Annual energy consumption (kWh) for low-charge refrigeration systems for selected locations

Location	Multiplex Air-Cooled	Low-Charge Multiplex Air-Cooled	Low-Charge Mutiplex Evap Condenser	Distributed Air-Cooled	Distributed Water-Cooled, Evaporative	Secondary Loop Evap. Condenser	Advanced Self-Contained Water-Cooled, Evaporative
Worcester, MA	904,500	850,000	791,600	785,700	802,200	821,600	983,700
Washington, DC	976,800	935,200	863,600	860,500	866,100	875,200	1,048,300
Memphis, TN	1,050,200	1,027,100	941,500	942,800	943,200	940,400	1,126,800
Los Angeles, CA	1,067,200	1,042,600	911,300	958,432	894,400	892,400	1,066,800

Angeles sites, while the distributed system showed lower energy consumption for the Worcester and Washington sites. The low-charge multiplex system showed less energy use than the multiplex baseline for all locations. Savings ranged from 2.2 to 6.0 percent and 10.4 to 14.6 percent for the low-charge multiplex with air-cooled and evaporative condensing, respectively.

The energy savings achieved by the distributed refrigeration system can be attributed to close-coupling of the compressors to the display case evaporators, operation of the scroll compressors at 60°F minimum condensing temperature, and the use of evaporative heat rejection with the fluid loop. Savings seen with the secondary loop system are due to close-coupling of the compressors and the chiller evaporator, subcooling produced by brine heating for defrost, and the use of evaporative condensing. The refrigeration energy of the secondary loop system was found to be less than that of the distributed system, but the added energy associated with brine pumping negated some of this advantage. The energy savings seen with the low-charge multiplex system were due to the ability of this system to operate at very low minimum condensing temperatures. The minimum condensing temperatures were 40 and 60°F for low and medium temperature refrigeration, respectively.

Total equivalent warming impact (TEWI) estimates were made for the refrigeration systems for operation in Washington, DC. These estimates are shown in Table ES-2. The distributed and secondary loop systems both showed significant reduction in TEWI, compared to the multiplex system.

Table ES-3 gives the estimated operating savings for the low-charge systems due to reduced refrigerant leakage. For this analysis, refrigerant costs of $1.75/lb and $7.75/lb were used for R-22 and R-404A, respectively.

Figure ES-2 and Table ES-4 show the analysis results for operation of the store refrigeration and HVAC. For all locations, the integrated system consisting of the distributed refrigeration system and the water-source heat pumps produce the lowest operating cost (combined cost for electric, natural gas, water, and refrigerant). Operating cost savings were estimated to be 11.1 to 19.2 percent when compared to multiplex refrigeration with conventional HVAC.

Table ES-5 shows the estimated payback of installed cost premium for several low-charge refrigeration systems. The low-charge multiplex system had an immediate payback, since no installed cost difference exists between the low-charge and baseline multiplex systems. The distributed system showed paybacks ranging from 3.4 to 7.0 years, while the secondary loop system showed paybacks of 8.3 to 16.8 years. These payback values are extremely sensitive to the installed cost premium for these systems, which is highly variable depending upon arrangements between the supermarkets and their equipment suppliers and installers. These cost differences are likely to be reduced for either the distributed or the secondary loop systems as more such systems are implemented.

Table ES-2. Total Equivalent Warming Impact (TEWI) for supermarket refrigeration

System	Condensing	Charge (lb)	Refrigerant	Leak (%)	Annual Energy (kWh)	TEWI (million kg of CO_2) Direct	Indirect	Total
Multiplex	Air-Cooled	3,000	R404A/	30	976,800	13.62	9.52	23.15
	Evaporative	3,000	R-22	30	896,400	13.62	8.74	22.36
Low-Charge	Air-Cooled	2,000	R404A/	15	935,200	4.54	9.12	13.66
Multiplex	Evaporative	2,000	R-22	15	863,600	4.54	8.42	12.96
Distributed	Air-Cooled	1,500	R404A	10	860,500	3.33	8.38	11.71
Distributed	Water-Cooled, Evaporative	900	R404A	5	866,100	1.00	8.44	9.44
Secondary Loop	Evaporative	500	R507	10	875,200	1.13	8.54	9.67
Secondary Loop	Water-Cooled, Evaporative	200	R507	5	959,700	0.23	9.36	9.58
Advanced Self-Contained	Water-Cooled, Evaporative	100	R404A	1	1,048,300	0.02	10.22	10.24

Results for site in Washington, DC – 15 year service life.
Conversion factor = 0.65 kg CO_2/kWh.
Multiplex – 33.3% R404A (low temperature), GWP = 3260; 66.7% R22 (medium temperature), GWP = 1700.
Distributed and Advanced Self-Contained – 100% R404A, GWP = 3260.
Secondary Loop – 100% R507, GWP = 3300.

Table ES-3. Estimated operating cost savings for reduced refrigerant leakage

System	Annual Leakage (lb) R-404A	R-507	R-22	Savings ($)
Multiplex - (R-404A/R-22)	300		600	
Multiplex - Low Charge	100		200	2,250
Multiplex - Low Charge Evap Cond	100		200	2,250
Distributed Air-Cooled	150			2,213
Distributed Water-Cooled, Evap	45			3,026
Secondary Loop Evap Condensing		50		2,988
Secondary Loop Water-Cooled, Evap		10		3,298
Advanced Self-Contained	1			3,367

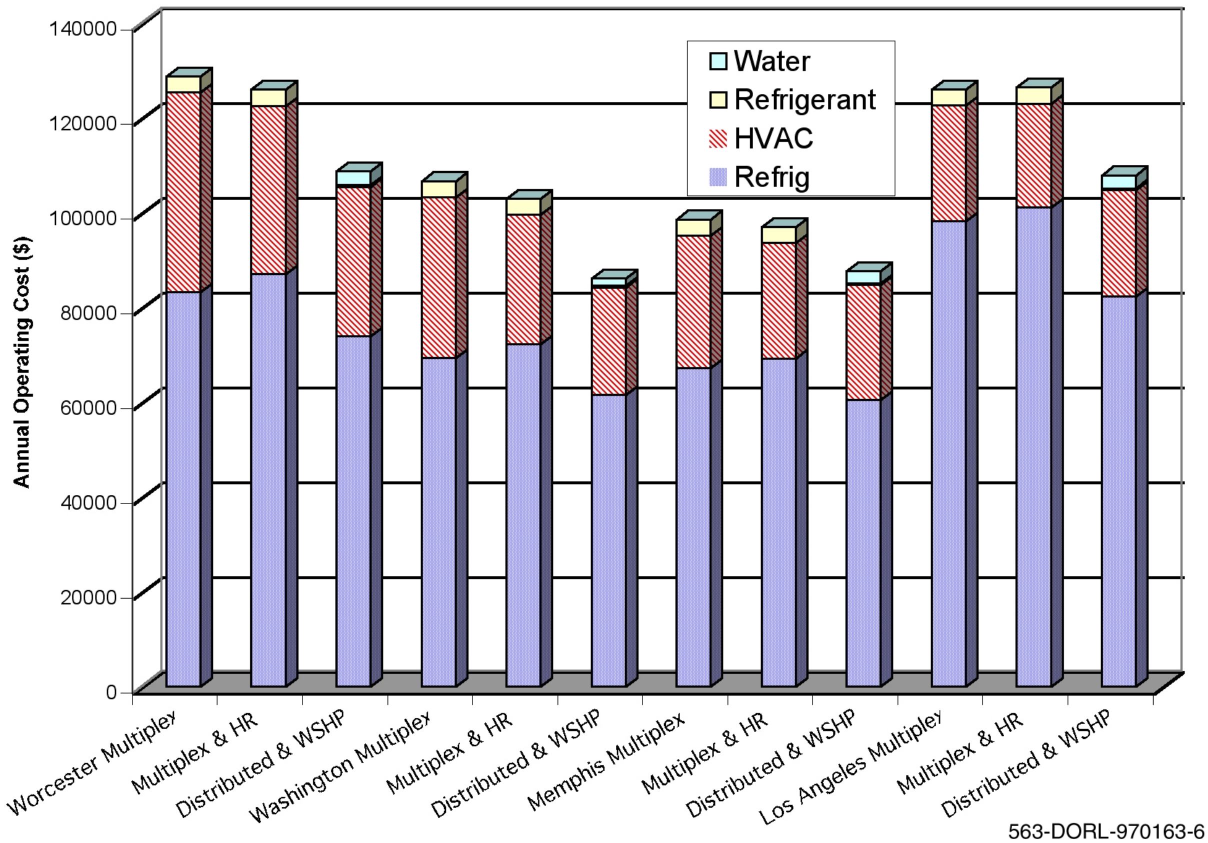

Figure ES-2. Operating cost of supermarket refrigeration and HVAC

Table ES-4. Annual cost savings achieved by distributed refrigeration and water-source heat pumps versus multiplex refrigeration and conventional HVAC

	Annual Operating Savings			
	with Heat Reclaim		Distributed Refrigeration and WS Heat Pumps	
Location	$	%	$	%
Worcester, MA	2,817	2.2	20,009	15.5
Washington, DC	3,732	3.5	20,469	19.2
Memphis, TN	1,568	1.6	10,929	11.1
Los Angeles, CA	-397	-0.3	18,079	14.4

Table ES-5. Estimated operating cost savings and payback for low-charge supermarket refrigeration (versus multiplex with air-cooled condensing)

Location	Low-charge Multiplex, Air-Cooled		Low-charge Multiplex, Evaporative		Distributed, Water-Cooled, Evaporative		Secondary Loop, Evaporative	
	$	Year	$	Year	$	Year	$	Year
Worcester, MA	7,264	0	11,176	0	10,977	5.5	9,153	16.1
Washington, DC	5,204	0	9,479	0	10,078	6.0	9,393	15.6
Memphis, TN	3,728	0	7,940	0	8,608	7.0	8,748	16.8
Los Angeles, CA	4,513	0	15,201	0	17,532	3.4	17,678	8.3

Table ES-6 shows the payback associated with distributed refrigeration and water-source heat pumps for combined operation of refrigeration and HVAC. The payback on cost premium for distributed refrigeration and water-source heat pumps was 4.2 years for Worcester, MA and Washington, DC, and 4.7 years for Los Angeles, CA. The payback for operation in Memphis, TN was 10.8 years. The lowest paybacks were seen for sites with large space heating loads. For these locations, the operation of the water-source heat pumps helped to reduce the combined payback.

The results seen in this investigation show that low-charge refrigeration systems can reduce energy and operating costs if properly designed and operated. Demonstration of these technologies by field testing, possibly in conjunction with water-source heat pumps for HVAC, will help develop best practices for these systems and also better quantify energy savings. This information will help to accelerate the use of low-charge refrigeration systems by the supermarket industry.

Table ES-6. Estimated payback for distributed refrigeration and water-source heat pumps

Location	Savings ($)		Payback (Year)	
	Refrigeration	Combined	Refrigeration	Combined
Worcester, MA	10,977	20,009	5.5	4.2
Washington, DC	10,078	20,469	6.0	4.2
Memphis, TN	8,608	10,929	7.0	7.8
Los Angeles, CA	17,532	18,079	3.4	4.7

1. INTRODUCTION

Supermarkets are the largest users of energy in the commercial sector. A typical supermarket (35,000 ft^2 of sales area) consumes on the order of 2 million kWh annually. Many larger superstores and supercenters also exist that can consume as much as 3 to 5 million kWh/yr.

One of the largest uses of energy in supermarkets is for refrigeration. Most of the product sold is perishable and must be kept refrigerated for storage and during display. Typical energy consumption for supermarket refrigeration is on the order of half of the store's total. Compressors and condensers account for 30 to 35 percent. The remainder is consumed by the display and storage cooler fans, display case lighting, and for anti-sweat heaters used to prevent condensate from forming on doors and outside surfaces of display cases.

Typical U.S. supermarket refrigeration systems today employ direct expansion air-refrigerant coils as the evaporators in display cases and coolers. Compressors and condensers are kept in a remote machine room located in the back or on the roof of the store. Piping is provided to supply and return refrigerant to the case fixtures. As a result of using this layout, the amount of refrigerant needed to charge a supermarket refrigeration system is very large. A typical store will require 3,000 to 5,000 lb of refrigerant. The large amount of piping and pipe joints used in supermarket refrigeration also lead to significant refrigerant leakage, which can amount to a loss of up to 30 percent of the total charge annually.

With increased concern about the impact of refrigerant leakage on global warming, new supermarket system configurations requiring significantly less refrigerant charge are now being considered. Advanced systems of this type include:

- Low-charge multiplex - Several refrigeration system manufacturers now offer control systems for condensers that limit the amount of refrigerant charge needed for the operation of multiplex refrigeration, which approach reduces the charge by approximately 1/3.

- Secondary loop – A secondary fluid loop is run between the display cases and a central chiller system. The secondary fluid is refrigerated at the chiller and is then circulated through coils in the display cases where it is used to chill the air in the case.

- Distributed – Multiple scroll compressors are located in cabinets placed on or near the sales floor. Scroll compressors are employed to minimize system noise in the sales area. The cabinets are close-coupled to the display cases and heat rejection from the cabinets can be done through the use of a glycol loop that connects the cabinets to a fluid cooler, in order to minimize refrigerant charge.

- Advanced self-contained – Self-contained refrigeration consists of compressors and condensers built into the display cases. An advanced version of this concept would used horizontal scroll compressors with capacity control, such as unloading, and water-cooled condensers with a water loop for heat rejection. The advanced self-contained refrigeration system would employ the smallest refrigerant charge.

While all of these systems use less refrigerant, energy use varies and can be much more than is now seen with centralized multiplex racks if the system design and component sizing does not take energy consumption into account. Examination of these systems on a total environmental warming basis (through TEWI) is needed to determine which has the least environmental impact.

The centralized refrigeration systems are mature in their technology and have been optimized in many ways, in terms of first cost and installation, reliability and maintenance, and energy use. At present, no regulations exist requiring reduction of refrigerant charge in supermarkets. For an advanced refrigeration system to replace centralized multiplex racks, such systems must offer an incentive for their use, such as reduced first cost, good return on investment for first cost, or lower operating and energy costs.

Many of the features now used to reduce energy consumption of the centralized systems could be employed with the advanced systems to increase their energy efficiency. Examples include reduced head pressure and mechanical subcooling. Compressor control strategies have been developed that can maintain suction pressure within a tight tolerance to a set point value. Similar control strategies that limit energy use have been implemented for condenser fans. Implementation of these energy-saving components and control strategies with the advanced, low-charge systems could lead to improved performance.

The advanced refrigeration systems also have inherent characteristics, which could lead to reduced energy consumption if these features can be utilized as much as possible. An example is that the secondary loop, distributed, and advanced self-contained refrigeration systems employ significantly shorter refrigerant suction lines (close-coupling), which mean that pressure drop and heat gain is much less than is seen with presently installed central refrigeration systems. Higher suction pressure and lower return gas temperature can translate into lower compressor energy consumption. Scroll compressors used in the distributed and advanced self-contained refrigeration system can operate at lower condensing temperature than reciprocating compressors, because scroll compressors have no suction valves. This feature could allow the distributed refrigeration system to operate at a much lower head pressure (i.e., floating head pressure), which has been shown previously (1-1) to be a method of significantly reducing refrigeration energy consumption. Low head pressure operation can also be obtained with the advanced self-contained refrigeration, if the compressors are equipped with capacity control. Low-charge multiplex refrigeration are operated at very low head pressure values, because of the improved charge control offered by these systems. Initial estimates suggest that incorporation of the energy-saving features in these advanced refrigeration systems could produce a reduction in refrigeration energy of as much as 10 to 12 percent of present use.

HVAC also represents a large portion of the energy use of a supermarket, on the order of 10 to 20 percent of the store total, depending upon geographic location. Refrigerated fixtures installed in a supermarket have a major impact on the store HVAC. For space cooling, the refrigeration removes both sensible and latent heat from the store, such that the sensible-to-latent load ratio is much smaller than is seen in most commercial buildings.

Also, because of the installed refrigeration, space heating is the dominant HVAC load. Reclaim of refrigeration reject heat for space heating has been done in past supermarket installations, but the amount of heat reclaimed is limited to desuperheating only due to operating considerations, which amounts to only 14 to 20 percent of the total amount of heat available.

A possible alternate approach to utilize refrigeration reject heat for space heating is through water-source heat pumps. The heat pumps can be installed in the glycol/water loop used for refrigeration heat rejection and use the refrigeration heat to provide space heating. This method offers several advantages, which are that a much larger portion of the reject heat can be reclaimed, and the condensing temperature and head pressure of the refrigeration system does not have to be elevated for the heat pumps to use the reject heat. Refrigeration energy savings achieved by low head pressure operation can be realized along with the energy benefits seen through heat reclaim.

The U.S. DOE initiated this research project, recognizing that advanced supermarket refrigeration systems have the potential of both environmental benefits and energy savings. The project involved investigation of low-charge refrigeration systems to determine configurations and designs that produced the largest energy savings compared to presently installed refrigeration systems. Along with examination of the refrigeration, integration of refrigeration and HVAC was also addressed to find an overall approach that provided maximum energy savings and operating cost reduction. Possible future project activities include installation of the combined refrigeration and HVAC system identified in a supermarket, and field testing and measurement of the performance of this advanced system.

Section 1 Reference

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2. DESCRIPTION OF SUPERMARKET REFRIGERATION SYSTEMS

A representative supermarket layout is shown in Figure 2-1. Refrigerated fixtures are located throughout the store, because of the large amount of perishable food products that are sold. These fixtures fall into two categories, which are display cases and walk-in storage coolers. The display cases are located on the sales floor and are designed to refrigerate food products while providing a place to merchandise them. Walk-in coolers are used to store food products during the time period between receiving the product and placing the product out for sale.

Refrigeration of the display cases and walk-in coolers is done through the use of direct expansion refrigerant/air coils located in each case or cooler. Refrigerant piping is provided to each coil to supply liquid refrigerant to the coil and remove evaporated refrigerant from the coil and return the gas to the refrigeration compressors.

The compressors are located in a machine room in a remote part of the store, such as in the back room area or on the roof. The system condensers are located either in the machine room, or more likely, on the roof above the machine room.

2.1 Refrigerated Display Cases

The purpose of refrigerated display cases in a supermarket is to provide temporary storage for perishable foods prior to sale. Most of the design characteristics and general shape and layout of display cases are based on marketing specifications and constraints. Display cases have been developed and refined for specific merchandising applications, and cases exist specifically for the storage and display of such items as frozen food, meats, fish, cheese, dairy products, and produce.

Despite the diversity of use and style, refrigerated display cases can be described as being of the following three general types (Figure 2-2):

- Tub (or coffin): The tub case is often used for the storage and display of frozen foods and meats. Tub cases operate at a very uniform temperature and require the least amount of refrigeration per foot of any display case type. The primary disadvantage of the tub is a low product storage volume per square foot of sales area taken up by the case.

- Multi-deck: The multi-deck case possesses the largest storage volume per square foot of floor area, because of the use of an upright cabinet and shelves. Refrigeration requirements are very high for multi-deck cases, including a large latent load portion due to the entraining of ambient air in the air curtain passing over the opening of the case.

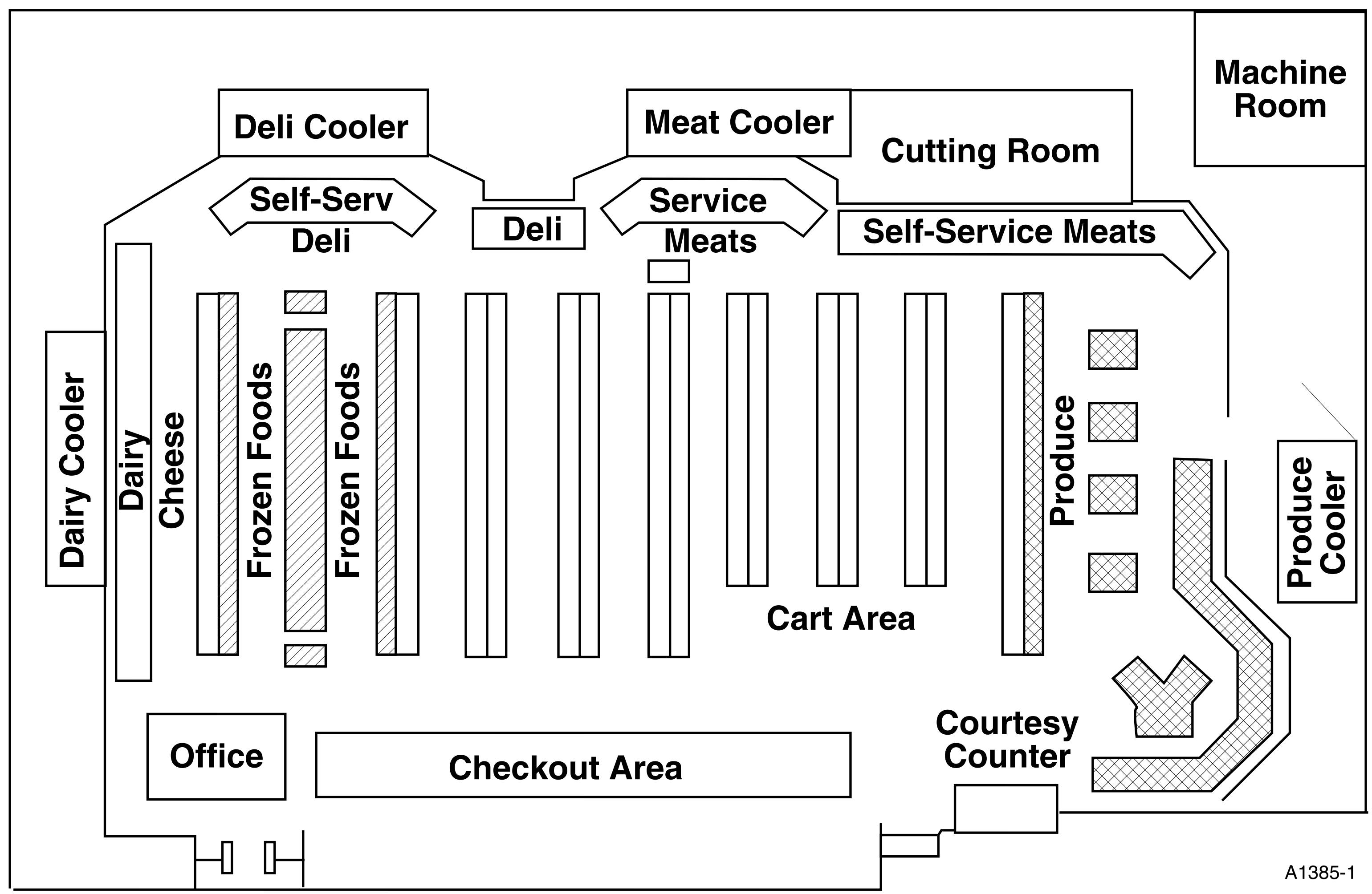
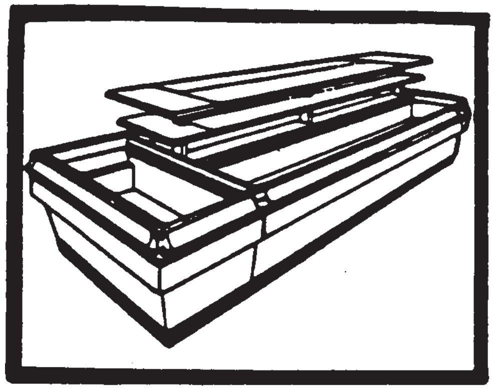
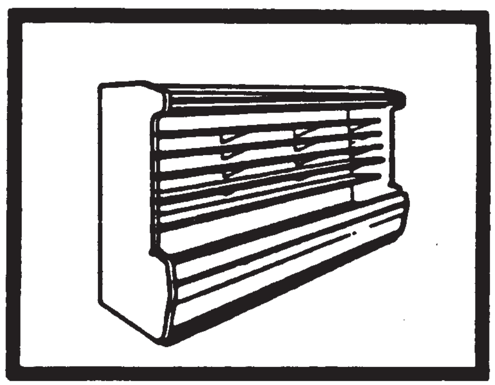


Figure 2-1. Layout of a typical supermarket

OPEN TUB



MULTI DECK - AIR CURTAIN



GLASS DOOR REACH-IN

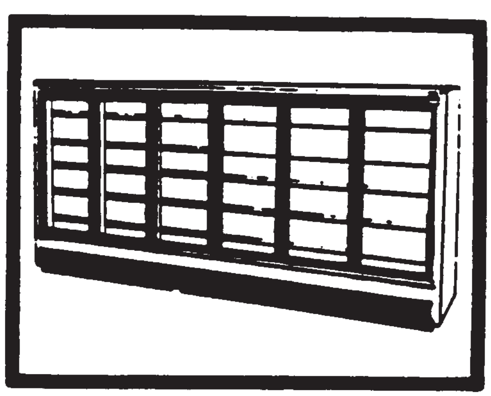


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Figure 2-2. Display case types employed in supermarkets

- Glass door reach-in: The reach-in case has glass doors over the opening of the case; these must be opened for product removal and stocking. Reach-in cases are used in supermarkets primarily for frozen foods, because of their ability to contain the cold refrigerated air, which reduces the “cold aisle” problem. The refrigeration loads associated with the glass door reach-in case are normally less than those for the multi-deck but greater than for the tub case. Glass door cases are, however, equipped with anti-sweat electric heaters in the doors to prevent fogging and decreased visibility of the product.

The design refrigeration load of a display case is influenced by its type and operating temperature. Table 2-1 presents the design refrigeration loads for several display case types, along with a description of the installed fan, light, and heater wattage.

2.2 Walk-In Storage Coolers

Walk-in storage coolers are used to hold food product prior to stocking in the display cases. One walk-in is normally associated with each type of food product, such as meat, dairy, produce, frozen food, etc. For meat and produce, the walk-in cooler is also used as a preparation area where the foods are cut, uncrated, or packaged prior to display. Walk-in coolers are fabricated from pre-insulated panels that are field assembled. Refrigeration is provided by one or more fan coils located on the ceiling of the walk-in. Walk-ins are equipped with doors large enough to allow pallet loads of product to be brought in or out. Doors are often left open so as not to impede the entry or removal of pallets. These open doors allow ambient air to pass into the cooler. This air infiltration is the largest component of the refrigeration load of the walk-in. Vinyl strips are sometime hung over walk-in doorways to provide a barrier to air movement and help reduce the amount of ambient air entering the walk-in.

2.3 Compressor Systems

Two compressor system types are now found in most supermarkets. They are the stand-alone and the multiplex, parallel compressor systems. Stand-alone systems are found in smaller

Table 2-1. Typical refrigeration and electric requirements for supermarket display cases

			Electric Requirements (W)*	
Display Case	Evaporator Temperature (ºF)	Refrigeration Load (Btu/h/ft)	Fans and Heaters	Lights
Frozen Food Tub	-25	600	595	-
Multi-Deck Dairy	15	1,800	440	345
Single-Deck Meat	15	550	190	-
Glass Door Reach-in	-25	560	1040	345

*For a case length of 12 ft

supermarkets or in stores that are more than 20 years old. The more predominate system is the multiplex system, which consists of multiple compressors piped in parallel on common skids and grouped by suction temperature. The operating characteristics of each of these compressor systems are described below.

2.3.1 Stand-Alone Refrigeration Systems

The characteristics of stand-alone refrigeration systems (Figure 2-3) are the use of a single compressor for each display case lineup or walk-in storage box; and skid-mounted construction with all necessary refrigerant piping, control valves, receiver, electrical components, and condenser mounted with the compressor on a base or skid.

The type of compressor employed for the stand-alone refrigeration system is typically a semi-hermetic reciprocating unit. The operation of the compressor is controlled through the use of a suction pressure control strategy in which the compressor suction pressure is held between set points (cut-in and cut-out pressures) by cycling the compressor on and off.

The individual condensers employed in the stand-alone refrigeration system consist of a plate fin-type coil that is air-cooled by a fan individually installed on each condenser. Two types of condenser control are employed. For systems where the condenser is mounted on the same skid as the compressor, the fan operates when the compressor is cycled on and is turned off when the compressor is cycled off. For remote condensers, head pressure control is employed where the condensing temperature is maintained at the desired value by fan cycling. The cycling is controlled through the use of a liquid line thermostat that is set at the desired minimum condensing temperature.

Standard operating practice for stand-alone compressor systems calls for a minimum condensing temperature of 90°F, which avoids short time periods of compressor operation, prevents short-cycling of the compressor. Minimal compressor cycling is desirable primarily to maintain a uniform air temperature at the display cases.

Two types of defrost are employed with the stand-alone refrigeration system. For the medium and high temperature fixtures, off-cycle defrost is commonly used, in which frost is allowed to melt from the display case evaporator. For the very low and low temperature display cases, electric defrost is employed; electric heaters melt the frost from the case evaporator.

2.3.2 Multiplex Refrigeration Systems

The term “multiplex” refrigeration refers to the use of multiple refrigeration compressors piped to common suction and discharge manifolds, and mounted on a skid as shown in Figure 2-4. The skid also contains all necessary piping, control valves, and electrical wiring needed to operate and control the compressors and the refrigeration provided to the display cases and walk-in coolers serviced by this particular compressor rack. The discharge gas from the compressors is piped to a remotely located condenser. Liquid refrigerant returning from the condenser is piped back to the compressor rack, where a receiver, liquid manifold and associated

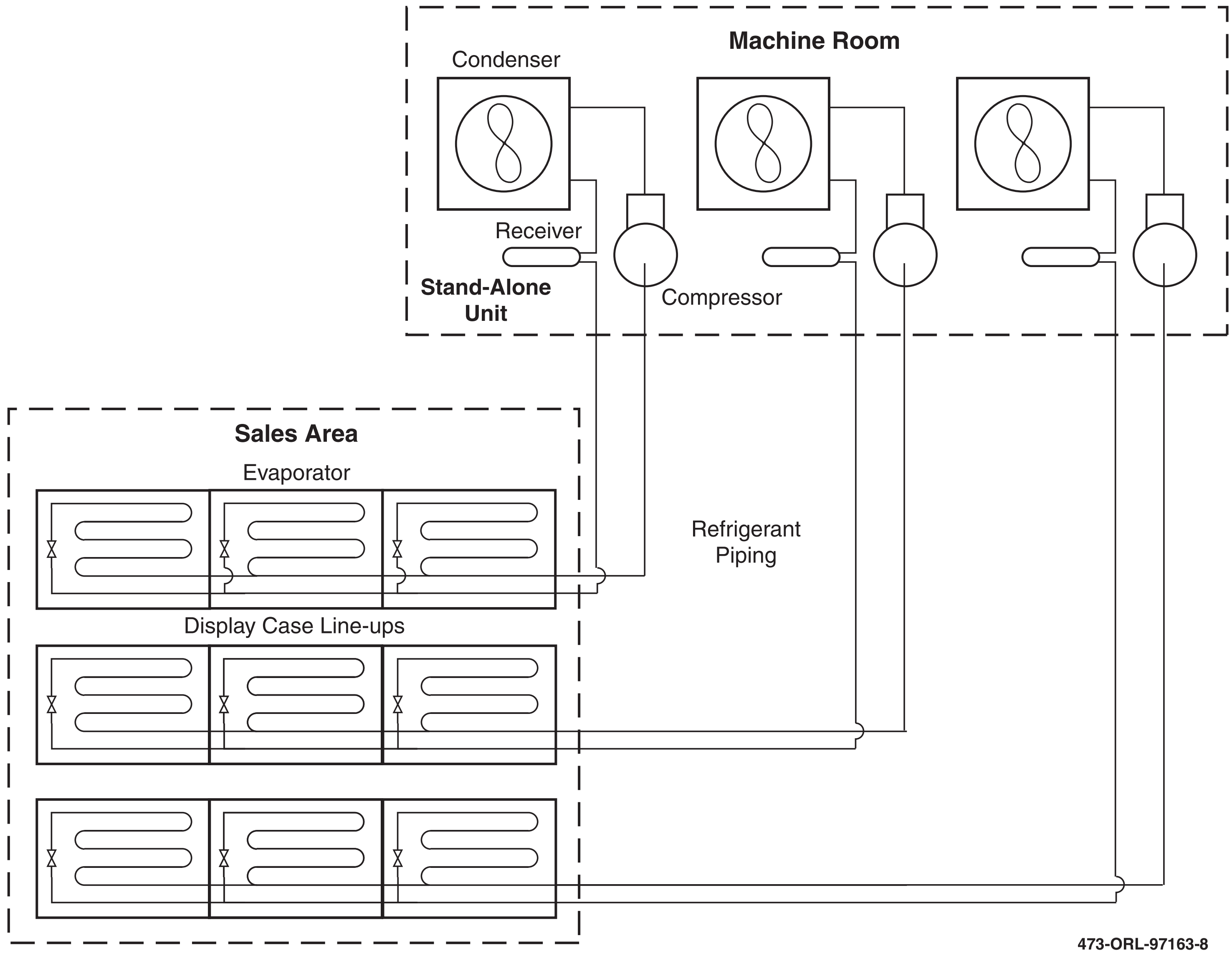

Figure 2-3. Elements of a stand-alone refrigeration system

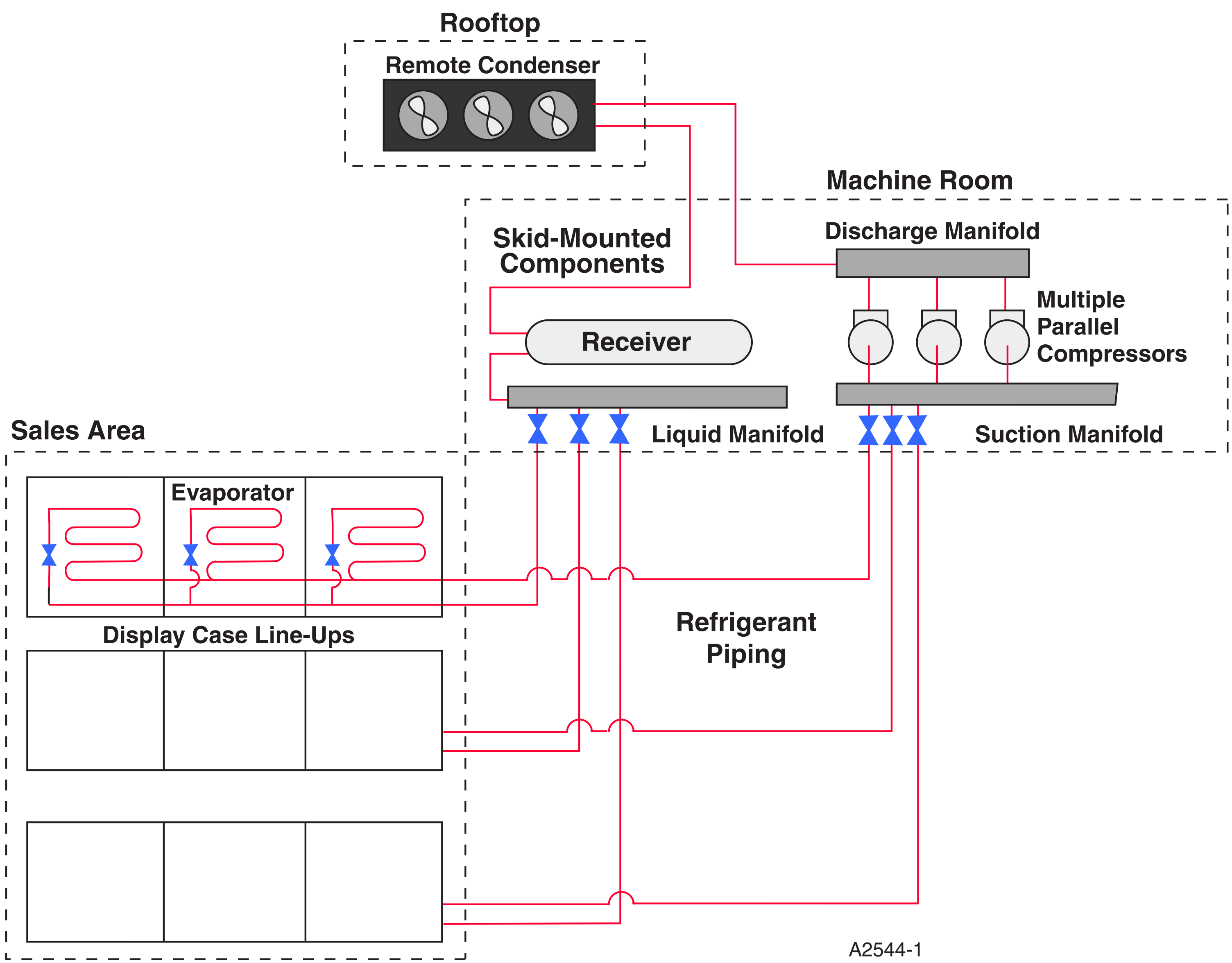

Figure 2-4. Diagram of a multiplex refrigeration system

control valves are located for distribution of the liquid to the cases and coolers. Each case or cooler circuit is piped with a liquid and suction return line that are connected to the liquid and suction manifolds located on the compressor skid. Valves used for control of each of these circuits are also located on the manifolds. The control valves employed consist of regulators to control suction pressure and solenoid valves used to control gas routing during defrost. The compressor is normally equipped with a number of pressure regulators used to control system head pressure, heat reclaim, and defrost. The rack will also have an oil separator in the discharge piping and an oil distribution system that will return oil to the compressors.

Typically 3 to 5 compressor racks will be employed to provide all refrigeration in the supermarket. The display cases and coolers are grouped and attached to the compressor racks based on required saturated suction temperature to maintain the desired case air temperature. A supermarket will have 1 or 2 low temperature racks to address all frozen food refrigeration requirements. The low temperature racks will typically operate at a -20°F SST. Refrigeration loads as low as -30°F, or as high as -10°F, will also be provided by the low temperature racks. In these situations, the suction manifold will be divided, and 1 or 2 compressors will provide the off-temperature refrigeration. The discharge of these "satellite" compressors will be piped to the common discharge manifold with the other low temperature loads so that a common condenser and liquid manifold can be used for all circuits on the rack. The remaining refrigeration circuits in the store are referred to as medium temperature and normally require a 20°F SST. Two or more compressor racks are needed to meet all medium temperature refrigeration requirements. Satellite compressors are also used for medium temperature loads requiring an SST significantly higher or lower than 20°F.

Multiplex systems commonly consist of three or four compressors that are sized such that operation of all compressors simultaneously can provide adequate capacity to meet the design refrigeration load. During off-design operation, the refrigeration load can be considerably less than the design value; at the same time, the refrigeration capacity of the compressors can increase with a decrease in ambient temperature and the condenser operating pressure. In this situation, the compressors operated by the multiplex system can be selected so that the capacity of the compressors closely matches the refrigeration load. The selection of, and the on-off cycling of the compressors is done based on suction pressure value measured at the compressor rack. The use of microprocessor-based controls allows more sophisticated control algorithms to be employed so that very close matching of the suction pressure and the set point value can be maintained with multiplex compressor systems.

Several advantages can be realized through the use of multiplexed compressor instead of single, stand-alone compressors for refrigeration. The matching of the capacity of the multiplexed compressors with the refrigeration load allows operation at the highest possible suction pressure to provide best compressor operating efficiency. In contrast, capacity control with a single compressor is accomplished by cycling within a larger suction pressure control band in order to prevent rapid on-off cycling that does not provide adequate air temperature stability at the display cases and shortens the service life of the compressor.

The use of multiplex compressors also allows operation at low head pressure because of the ability of the multiplex systems to match capacity and load continuously. The lowest head pressure seen in the operation of multiplex systems employing reciprocating compressors corresponds to a saturated discharge temperature (SDT) of 70°F, which is the lowest condensing temperature that is recommended for the operation of such compressors. Operation with low head pressure has been shown to produce energy savings of about 10 percent in compressor energy (2-1) when compared to a system operating at a minimum condensing temperature of 95°F. If scroll compressors are employed in a multiplex system, operation at lower condensing temperature than 70°F is possible. The lowest condensing temperatures seen for scroll compressors are 40 and 60°F for low and medium temperature refrigeration, respectively. These minimum values are set by the requirement that the compressor maintain a minimum pressure difference between suction and discharge in order to maintain proper oil flow for lubrication.

Other energy saving features associated with multiplex refrigeration is the use of mechanical subcooling for low temperature refrigeration. Figure 2-5 shows a piping diagram for mechanical subcooling. The liquid refrigerant flow for the low temperature system is passed through a heat exchanger where the liquid is cooled by refrigeration provided by a medium temperature refrigeration rack. The subcooling load is part of the total refrigeration load of the medium temperature rack and is shared by the compressors on the rack. Mechanical subcooling typically produces energy savings of approximately 8 percent for low temperature compressor energy consumption (2-1).

Multiplex refrigeration systems also often employ hot gas defrost, particularly for low temperature refrigeration. The piping arrangement for hot gas defrost is shown in Figure 2-6. Discharge gas is directed to the circuit requiring defrost and is passed down the suction line piping. The melting of the frost condenses the gas and the liquid is piped back to the liquid refrigerant manifold. Hot gas defrost replaces electric heaters in the display cases which heat the case air to remove frost from the coil. Savings obtained by the use of hot gas defrost is on the order of 4 percent of total compressor energy (2-1).

Reclaim of refrigeration reject heat has been done with multiplex refrigeration systems. Heat reclaim has been used for both water and space heating. Water heating is most prevalent and consists of a water tank equipped with a heat exchanger. The discharge gas from one of the low temperature racks is piped to the heat exchanger where the gas is desuperheated to heat the water in the tank.

Figure 2-7 shows the piping diagram for heat reclaim for store space heating. Operation of the heat reclaim circuit is controlled by a 3-way valve mounted in the discharge piping. The valve is actuated by a thermostat normally located in the store sales area. The discharge gas from the rack is routed to a coil mounted in the HVAC ducting, usually at the air handler. At the coil, the refrigerant is desuperheated and partially condensed by the heating of circulated store air. The refrigerant is piped back to the condenser where the remainder of the condensing occurs and the liquid is returned to the multiplex rack.

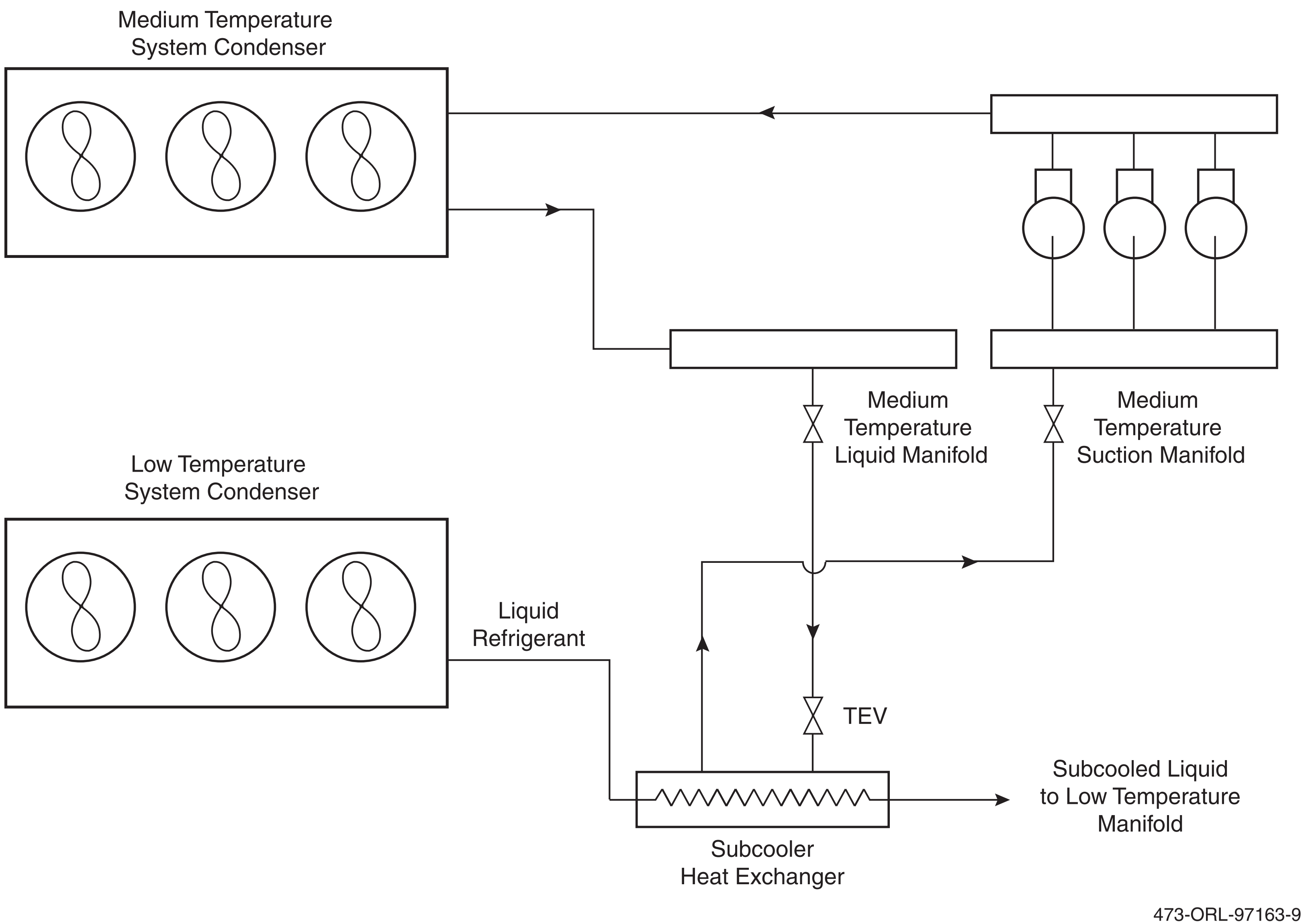

Figure 2-5. Mechanical subcooling of low temperature refrigerant

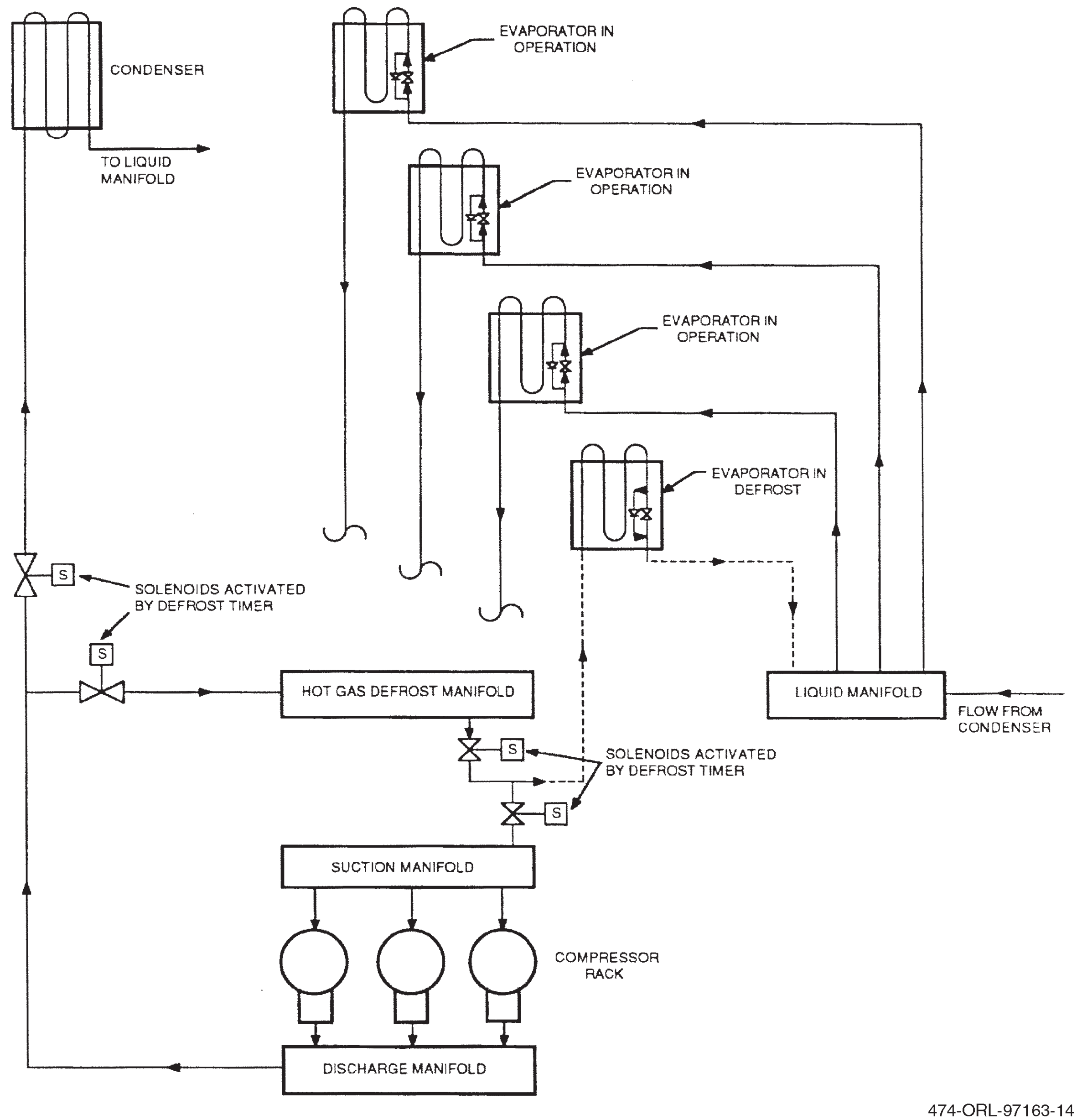

Figure 2-6.   Schematic of hot gas defrost

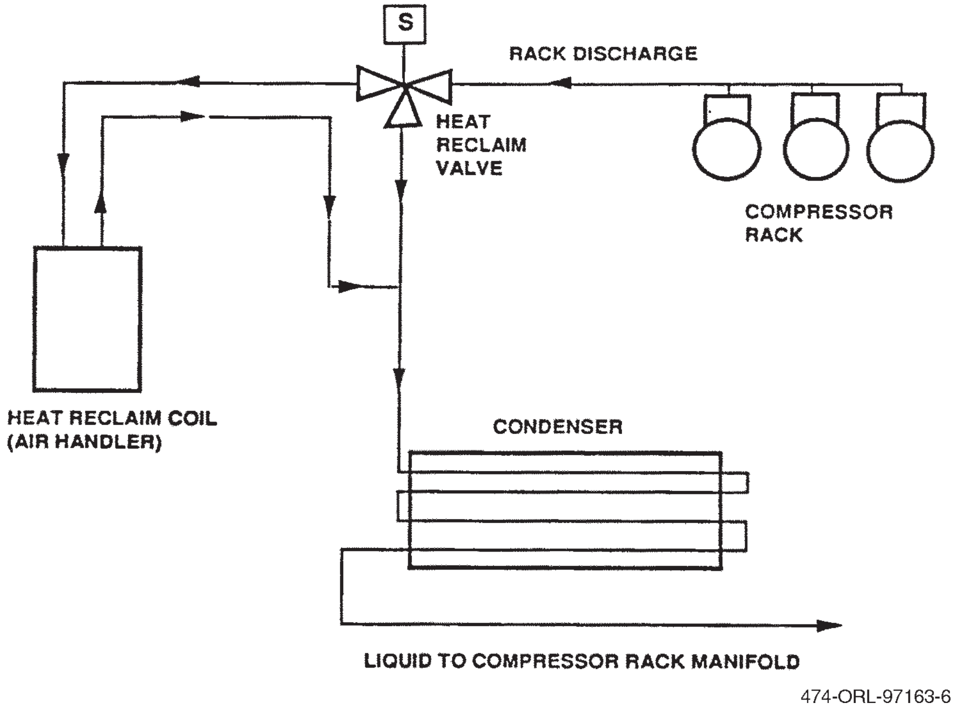

Figure 2-7. Refrigeration heat reclaim schematic

2.4 Compressors

The semihermetic reciprocating compressor is the most common compressor type used in supermarket refrigeration. In this type of compressor, the mechanical components of the compressor and the electric drive motor are contained within a common housing. The refrigerant gas returning to the compressor suction passes over the motor, providing cooling. The compressor housing is constructed of bolted sections, which can be separated to provide access for service and repair, thus the term "semihermetic."

Semihermetic screw compressors have made some in-roads into supermarket refrigeration, primarily because the cost of such machines has dropped due to the use of advanced manufacturing methods. The reliability of screw compressor is accepted as being higher than that of reciprocating compressors, because no valves are employed and the screws are impervious to liquid refrigerant slugging. In terms of performance, screw compressors require higher power input than reciprocating compressors at equivalent load and operating conditions. Performance of the screw compressor can be increased by the use of an economizer, which provides liquid subcooling by evaporation of a portion of the liquid refrigerant in a heat exchanger that provides cooling to the remaining refrigerant liquid. The refrigerant vapor generated is injected in the screw compressor at a mid-screw location where the pressure is higher than the suction pressure of the compressor. The use of an economizer is akin to mechanical subcooling where medium temperature refrigeration is used to subcool the refrigerant liquid used for low temperature refrigeration.

Scroll compressors have been recently introduced in sizes appropriate for supermarket refrigeration. The primary advantage of scroll compressors is very quiet operation, making them suitable for use in the sales area. Scroll compressors are also tolerant to liquid slugging due to the use of moveable scroll elements, which allow the scrolls to separate if a liquid slug passes through. Scroll compressors show lower efficiency than reciprocating units for both medium and low temperature refrigeration. The loss in efficiency is caused primarily by re-expansion loss at the discharge, because no discharge valve is employed. Scroll compressors offer other energy saving possibilities. Mid-scroll injection of vapor can be used as a form of economizing to subcool refrigerant liquid. This procedure is similar to that described previously for screw compressors. Scroll compressors can be operated at lower condensing temperature than reciprocating compressors, because scroll compressors employ no valves. Lower floating head pressure values can be employed. The only limit is that a minimum pressure ratio, of approximately 2 to 1, must be maintained for proper operation of the scroll elements. For supermarkets, the systems in question are the higher end medium temperature such as produce or the meat prep area which can have an evaporator temperature as high as 35°F. This temperature will limit the minimum condenser temperature to about 60°F for refrigerants such as R-404A or R-507.

Scroll compressors used for multiplex systems, are used exclusively for distributed refrigeration for noise reduction in the sales area, and could possibly be used for self-contained, if compressors with capacities that match display case loads are available. These small capacity scroll compressors should also be equipped with capacity control, such as unloading to allow low head pressure operation.

2.5 Condensers

The most common type of condenser used in supermarket refrigeration is air-cooled. The reason for this is that air-cooled condensers require the least maintenance and have been shown to operate reliably in the non-operator environment of supermarket refrigeration.

Finned coil construction with 8 to 10 fins/in. and multiple fans are used. Typical face velocity requirements are on the order of 500 fpm. Fan motor sizes associated with air-cooled condensers are on the order of 1/2 to 1 hp. Multiple fans are employed to ensure air flow through the entire coil face. On/off fan cycling is used as a means to control condensing temperature and to reduce fan energy at ambient temperatures less than design.

Evaporative condensers are also used in some supermarkets, primarily in drier climates where a substantial difference in dry-bulb and wet-bulb temperatures exist. The evaporative condenser consists of a tube bundle, a fan for air flow, and a water sump and pump system used to spray water over the tube bundle. The refrigerant vapor is passed through the tube bundle where heat is removed and the refrigerant is condensed. The resulting condensing temperature can be close to the ambient wet-bulb. Evaporative condensers require less air flow than air-cooled condensers of equivalent rejection capability and can, therefore, be operated at a lower minimum condenser temperature without a fan energy penalty.

Water treatment and consumption are major issues that prevent more use of evaporative condensers in supermarkets. Water treatment is needed because of the evaporation of the water, which tends to concentrate dissolved minerals and other solids in the water, which eventually will precipitate and form deposits on tube surfaces. Exposure of the water to air also causes biological growth within the evaporative condensers in the form of algae and slime. Treatment of evaporative condenser water consists primarily of "blowdown" in which a fraction of the water is discharged to the drain and replaced with fresh water. The discharged water will carry away excess minerals and solids, which prevents solids precipitation. Biocides, such as chlorine, are also added to the water by automatic drip systems to prevent biological growth.

Water-cooled condensers are used primarily in urban stores where easy access to the outside for condensers does not exist. Stand-alone compressors or compressor racks are equipped with water-cooled condensers and glycol/water loops are used to circulate liquid between the condensers and a fluid cooler. The water/glycol loop is closed and is not exposed to the air so that algae and other biological formation do not occur. The only water treatment needed consists of maintenance of pH and the addition of corrosion inhibitors in the loop, which need be done on a very limited bases at time intervals of one year or greater.

The fluid cooler can be either air or evaporatively cooled. Air-cooled units require less maintenance since no water is exposed to the air. The use of a water loop with a dry fluid cooler results in higher condenser temperatures than seen with air-cooled condensing, because of the added temperature difference needed to transfer heat to the water. The dry fluid cooler must operate at a temperature higher than ambient dry-bulb in order to reject heat. Air flow and fan power requirements are similar to those seen with air-cooled condensers. Evaporatively cooled fluid coolers operate similarly to evaporative condensers and reject heat at a temperature close to the ambient wet-bulb. Water temperatures less than ambient dry-bulb can be obtained, which can reduce the condensing temperature to a value similar to that achieved with air-cooled condensing. Fan power requirements are of the same order as seen with evaporative condensers. Evaporative fluid coolers employ the same water treatment as is used with evaporative condensers.

Condenser control consists of maintaining a minimum condensing temperature based on a pressure reading seen at the condenser inlet. The condenser fans are cycled based upon a set point value of this pressure.

Pressure regulators are sometimes used to regulate the pressure of the condenser, often in combination with other control valves for functions such as hot gas defrost or heat reclaim. The regulator restricts liquid flow from the condenser, which builds the liquid level, causing the pressure to rise. Some subcooling of the liquid can occur when the flow is regulated in this fashion. Fan cycling is often used in conjunction with the pressure regulator for condenser pressure control.

Section 2 Reference

2-1. Walker, D.H., and G.I. Deming, *Supermarket Refrigeration Modeling and Field Demonstration,* Foster-Miller, Inc., EPRI Report No. CU-6268, Electric Power Research Institute, Palo Alto, CA., March, 1989.

3. ADVANCED REFRIGERATION SYSTEMS

The advanced supermarket refrigeration systems described here were designed specifically to reduce the amount of refrigerant needed for operation. Four such systems were identified consisting of low-charge multiplex, distributed, secondary loop, and advanced self-contained. The low-charge multiplex system is designed to limit the refrigerant charge to the minimum needed for operation. The distributed refrigeration system has the compressors located close to the display cases in cabinets on or near the sales area. Heat rejection is done with either air-cooled condensers located on the store roof above the compressor cabinets or through the use of a fluid loop that connects the condensers in the compressor cabinets with a fluid cooler on the roof of the supermarket. The secondary loop system employs a secondary fluid loop to refrigerate the display cases. A central chiller in a machine room, away from the sales area, is used to cool the secondary fluid loop. Heat rejection for secondary loop systems can be done with air-cooled or evaporative condenser, or by a fluid loop. The advanced self-contained approach employs compressors in 1 to 3 display cases and uses a fluid loop, like the distributed system, for heat rejection.

3.1 Low-Charge Multiplex Refrigeration

Several refrigeration system manufacturers now offer control systems for condensers that limit the amount of refrigerant charge needed for the operation of multiplex refrigeration. Figure 3-1 shows an example of such a control approach. A control valve is used to operate a bypass from the condenser liquid line in order to maintain a constant differential between the high and low pressures of the system. The refrigerant liquid charge is limited to that needed to supply all display case evaporators. No added liquid is needed for the receiver, which is included in the system, primarily for pump-down during servicing. All refrigerant liquid bypassed is expanded and evaporated through heat exchange with the discharge manifold. The resulting vapor is piped to the suction manifold for recompression and return to the condenser. The use of this control approach reduces the charge needed by the refrigeration system by approximately 1/3.

The control of the liquid charge by this method offers some energy-saving potential, because it has been found that compressors can be operated at very low head pressures when this control method is employed. The minimum condensing temperature values suggested for this low-charge system are 40 and 60°F for low and medium temperature refrigeration, respectively (3-1).

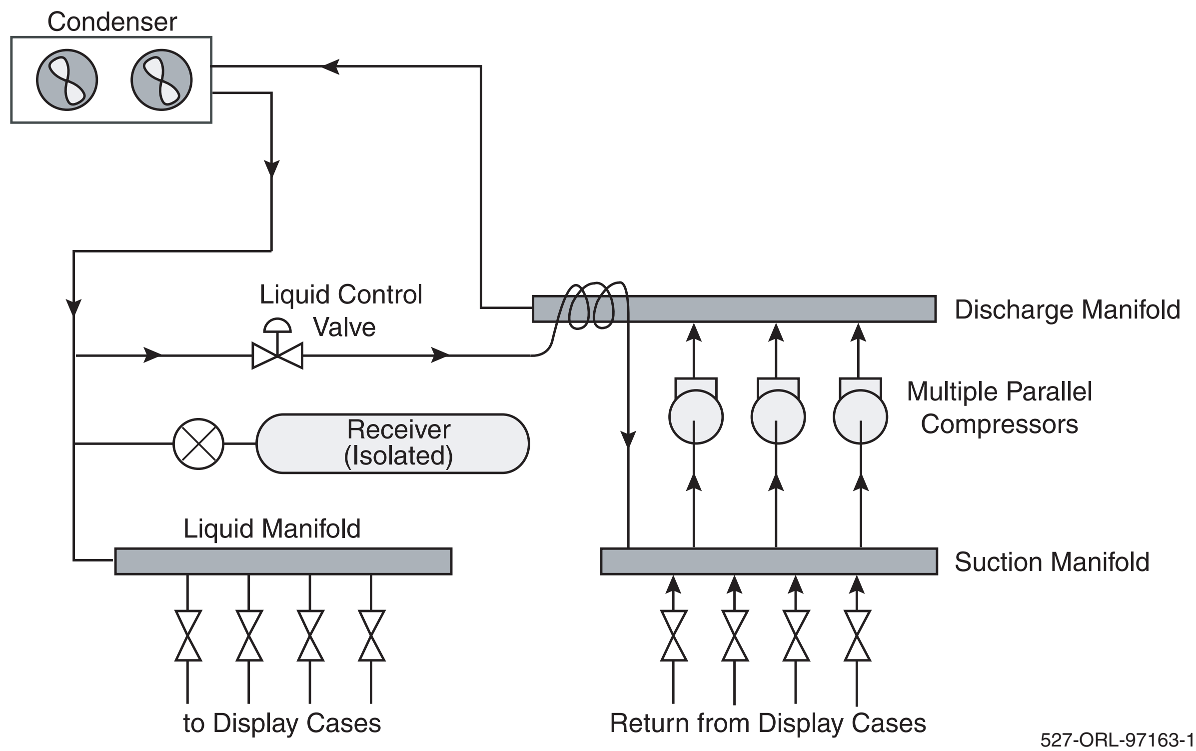

Figure 3-1. Piping diagram for the low-charge multiplex system

3.2 Distributed Refrigeration

Figure 3-2 shows a diagram of the distributed refrigeration system. Cooling of the display cases is provided by direct expansion coils as is done in present supermarkets. The difference is that the long lengths of piping needed to connect the cases with the compressor racks have been eliminated. The compressors are located in cabinets that are close-coupled to the display case lineups. The cabinets are placed either at the end of the case lineup or, more often, behind the cases around the perimeter of the store.

Figure 3-3 shows typical locations of the compressor cabinets in a supermarket. The cabinets are located within the store to provide refrigeration to a particular food department, such as meat, dairy, frozen food, etc. With this arrangement, the saturated suction temperature (SST) employed for each rack closely matches the evaporator temperature of the display cases and walk-in coolers. This is not always the situation seen with multiplex, since a single rack will often provide refrigeration to display cases with three or four different evaporator temperatures. The multiplex system must operate at a SST value that will satisfy the temperature requirements of all display cases connected. The better temperature matching seen with distributed refrigeration benefits the energy consumption of the system.

The refrigerant charge requirement for the distributed system is much less than is seen for multiplex refrigeration. The reduction in charge is due to the shortening of the suction and liquid

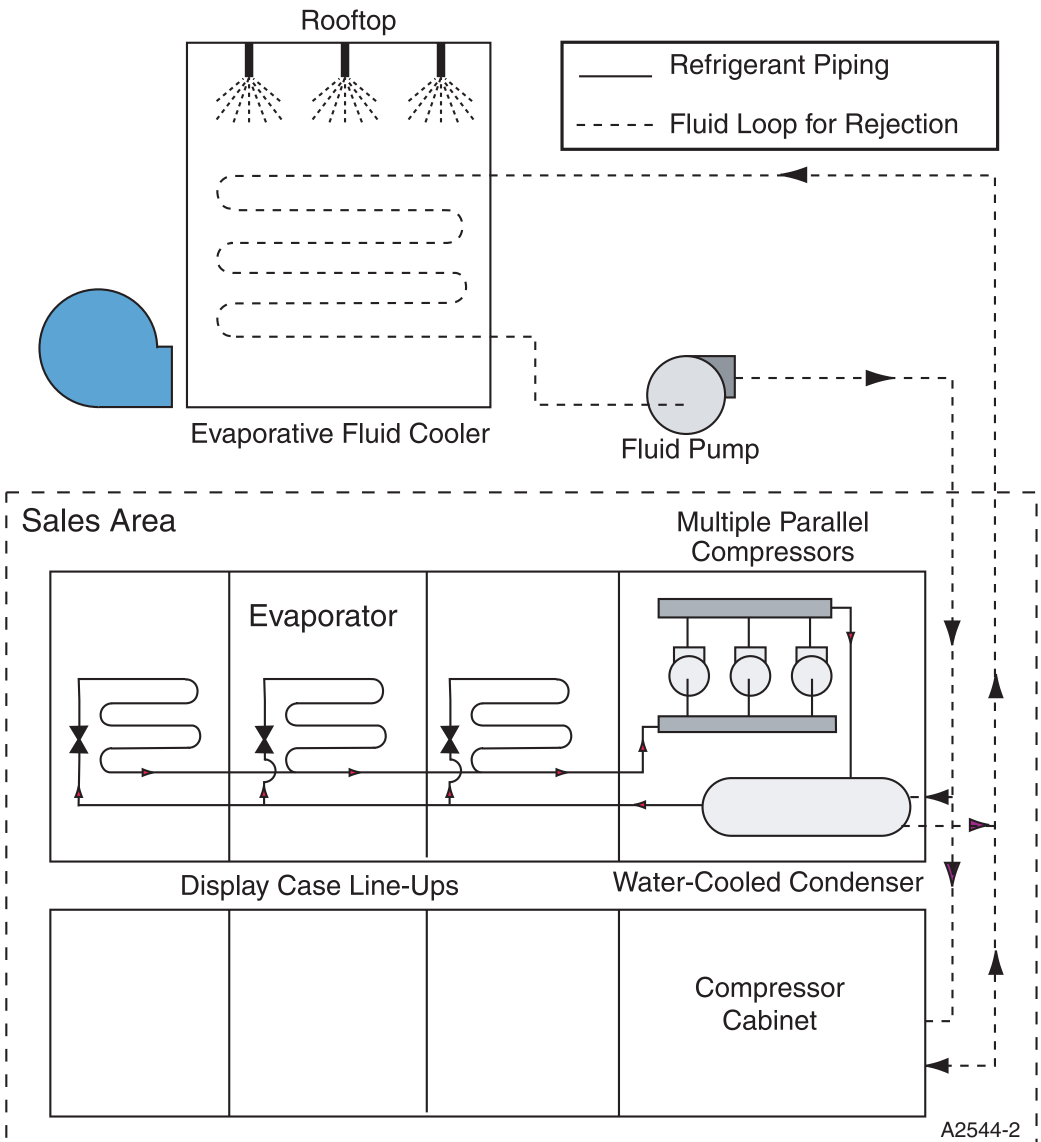

Figure 3-2. Description of the distributed refrigeration system

lines to the display cases. And the elimination of the refrigerant heat rejection piping to a remote condenser. The refrigerant charge associated with each compressor cabinet is about 90 lb. Since 9 to 10 cabinets are needed to provide all refrigeration in a supermarket, the total refrigerant charge is 810 to 900 lb.

Each compressor cabinet is similar to a multiplex rack. All necessary electrical and piping connections are provided within the cabinet, such that the only field connections are the refrigerant liquid and suction lines, fluid inlet and outlet piping for heat rejection, and electric service wiring. Multiple compressors are employed, which are piped in parallel so that multiplex operation can be used for capacity control of the refrigeration. Different size compressors are employed so that a mix and match approach can be used to maintain the desired suction pressure

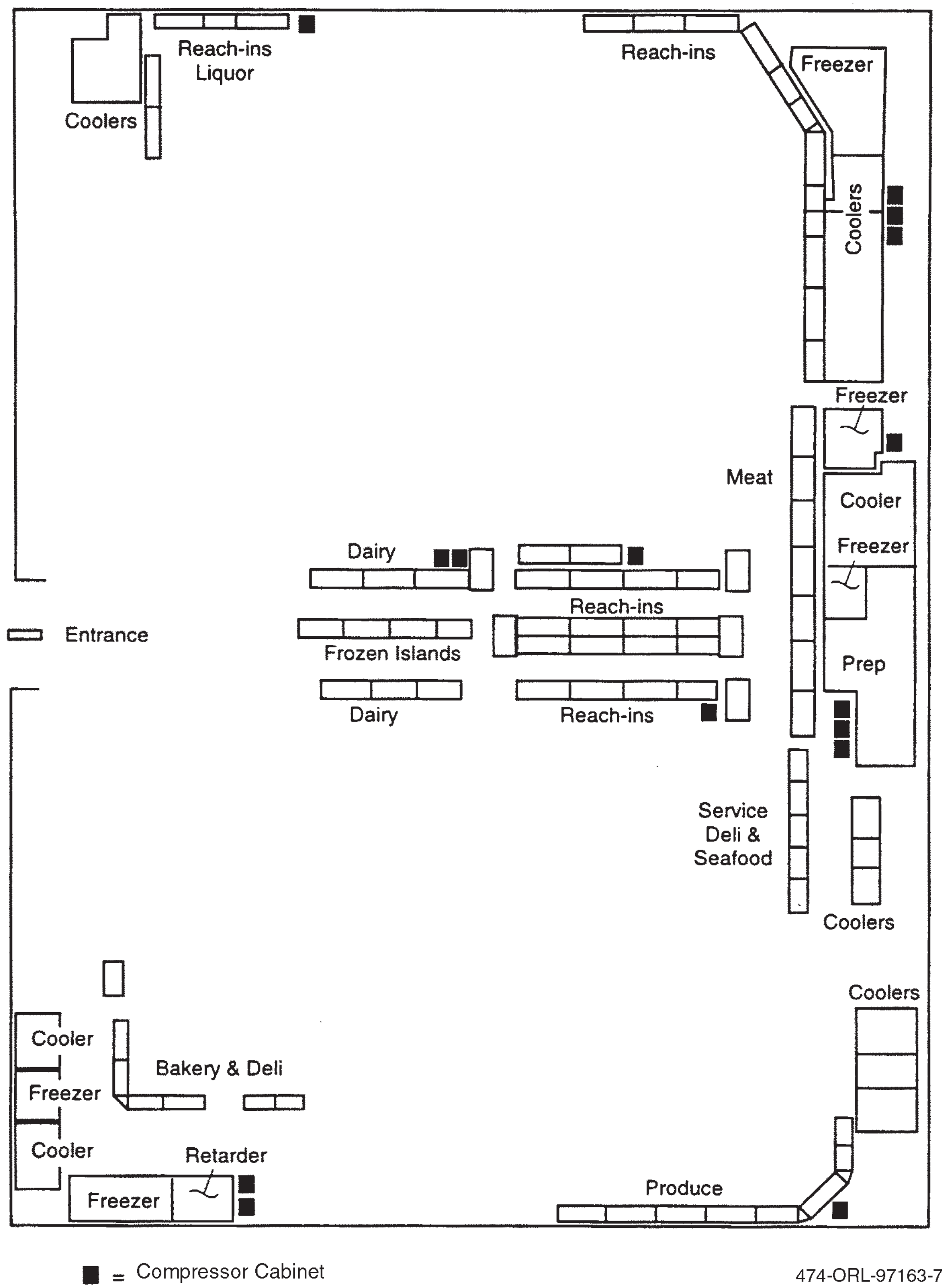

Figure 3-3. Supermarket layout using a distributed refrigeration system

set point. Usually, 3 to 5 compressors are installed in each cabinet. The cabinets are equipped with discharge and suction manifolds for parallel piping of the compressors. The suction manifold can be divided so that multiple suction temperatures are provided from a single cabinet.

The distributed refrigeration system cabinets can be equipped with hot gas defrost. The method used consists of a 3-pipe approach where a hot gas manifold is fed from the discharge manifold, and each evaporator in the system is equipped with a hot gas line that is controlled by a solenoid valve. During defrost, the hot gas flows down the hot gas line of the particular case being defrosted. The resulting condensed liquid refrigerant is returned to the cabinet by the liquid line.

The distributed refrigeration system employs scroll compressors, because of the very low noise and vibration levels encountered with this type of compressor. These characteristics are necessary if the compressor cabinets are located in or near the sales area. The scroll compressors offer several features that can produce significant energy savings for refrigeration operation. The compressors have no valves, and, therefore, can be operated at significantly lower condensing temperature. The lowest condensing temperature possible is at a suction-to-discharge pressure ratio of 2, which means for supermarket systems that the lowest condensing temperature possible is on the order of 55 to 60°F.

The scroll compressors are capable of mid-scroll injection of refrigerant vapor, which can be used to subcool liquid refrigerant. The subcooling is done using a heat exchanger in the liquid line that is mounted in the compressor cabinet. A portion of the liquid is taken from the liquid line and is expanded into the exchanger to provide cooling for the remaining liquid. The vapor generated at the heat exchanger is piped to the scroll compressors at their injection ports. The performance obtained by mid-scroll injection subcooling is determined from manufacturer's performance data for the scroll compressor when subcooling is applied. The amount of vapor injection is limited and the resulting subcooled liquid temperature will vary from 15 to 20°F below the liquid temperature at the outlet of the condenser.

The close-coupling of the display cases to the distributed refrigeration cabinets has other ramifications to energy consumption. The shorter suction lines mean that the pressure drop between the case evaporator and the compressor suction manifold is less than that seen with multiplex systems, which means that the SST of the cabinet will be close to the display case evaporator temperature. Typically, the SST of multiplex racks will be 2 to 4°F less than the case evaporator temperature. SST values for the distributed system will be about 1 to 2°F less than the case evaporator temperature. The shorter suction lines also mean that less heat gain to the return gas is experienced. The cooler return gas has a higher density and results in higher compressor mass flow rates, which means that less compressor on-time is needed to satisfy the refrigeration load. The return gas temperature rise seen for the distributed refrigeration system is normally on the order of 5 to 15°F, depending on the distance between the cabinet and the display cases and the evaporator temperature of the display cases. A greater return gas temperature rise is seen in low temperature systems than is seen in medium temperature. In comparison, the return gas temperature rise seen with multiplex systems falls between 40 and 65°F, due to the longer length of suction lines employed. The liquid temperature can also be affected detrimentally by line length if subcooling is employed. Heat gain to a subcooled liquid line will result in a rise in liquid refrigerant temperature before reaching the display cases.

The layout of a glycol/water loop for heat rejection for a distributed refrigeration system is shown in Figure 3-4. A central pump station is provided that contains the circulation pump and all valving needed to control fluid flow between the cabinets and the fluid cooler. Flow to and from the fluid cooler and pump station is provided by single inlet and outlet pipes sized for the entire system flow. The flow to each of the compressor cabinets is branched from these central supply and return pipes. The flow rate through a compressor cabinet is on the order of 10 to 30 gpm, depending upon the amount of refrigeration provided by the cabinet. The total flow in the heat rejection loop is about 300 to 350 gpm. This flow rate is continuous. Flow to each cabinet is controlled by manual balancing valves that are set at installation to ensure proper flow to each cabinet.

The lowest ambient temperature anticipated during operation determines the glycol concentration of the loop. The typical value of concentration is 25 to 33 percent, which is appropriate for ambient temperatures of 0 and –20°F, respectively.

3.3 Secondary Loop Refrigeration

Figure 3-5 shows the elements of a secondary loop supermarket refrigeration system. The difference between the secondary loop and direct expansion systems is that the display cases and storage coolers are refrigerated by a chilled secondary fluid loop. The secondary fluid is piped to

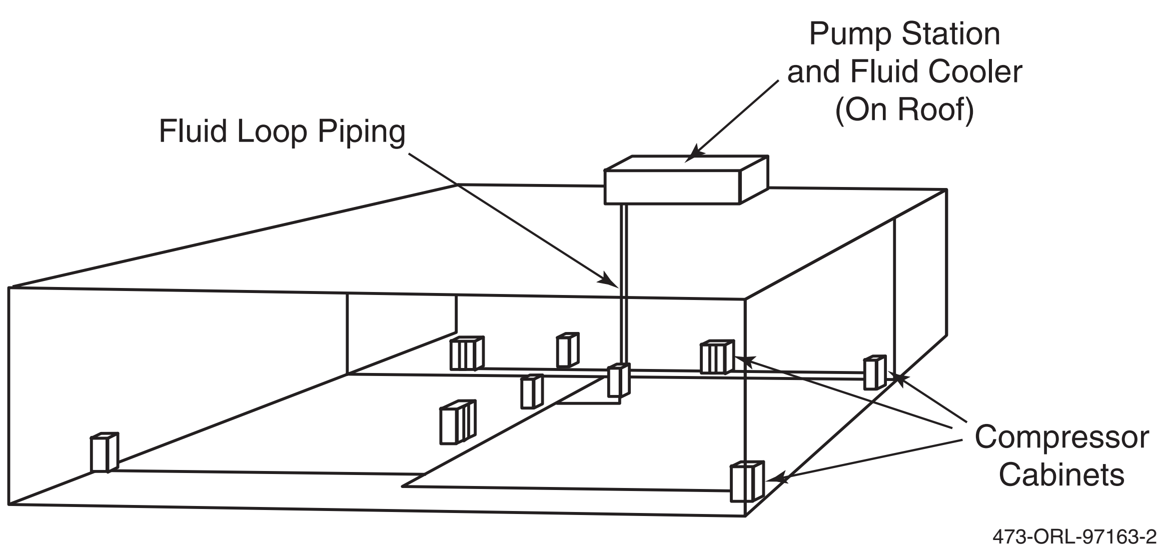

Figure 3-4. Rejection loop pipe layout for the distributed refrigeration system

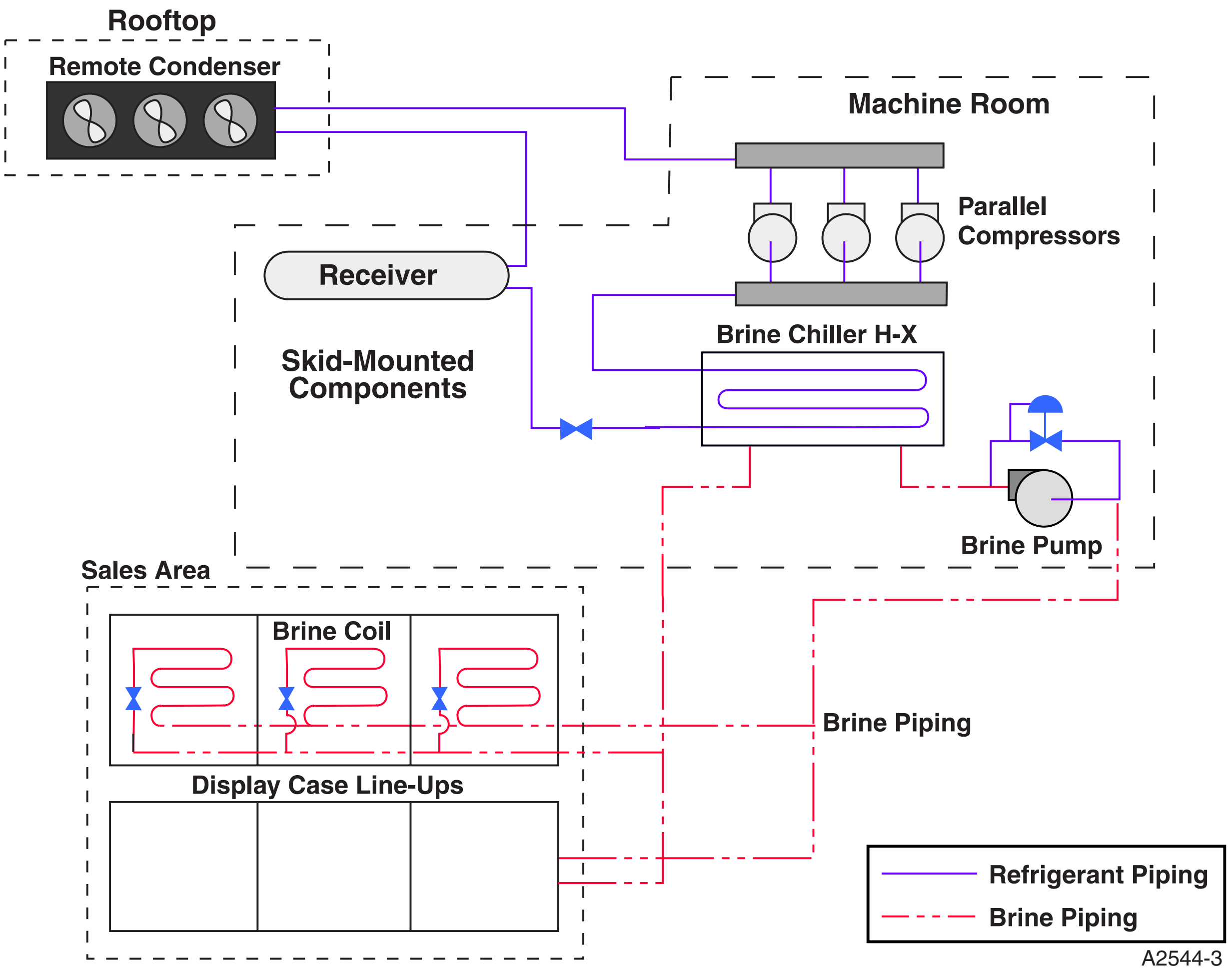

Figure 3-5. Elements of the secondary loop refrigeration system

heat exchanger coils in the display cases through which the case air is passed in order to provide cooling. The secondary fluid is chilled by a central system located in a machine room away from the sales area. The chiller is similar in configuration to the multiplex compressor racks. The only exception is that an evaporator is included with the chiller to provide refrigeration to the secondary fluid. The chiller package may also contain the pumps for the fluid loop. The multiple compressors are piped in parallel, and the suction of the compressors removes refrigerant vapor from the evaporator. The discharge of the compressors is to a common manifold and the discharge gas is piped to a remote condenser, normally located on the roof above the machine room. Secondary loop refrigeration can be configured to operate with air-cooled, evaporative, or water-cooled condensers. The use of evaporative condensing can produce the lowest average condensing temperature with lower fan energy than seen with air-cooled condensers.

The secondary loop refrigeration can be configured to operate with 2 to 4 separate secondary fluid loops and chiller systems. In the 2-loop configuration, the secondary fluid temperatures are 20 and -20°F for medium and low temperature refrigeration, respectively. As a consequence of the use of only two temperatures for refrigeration, all display cases and storage coolers must operate with these two temperatures. For this reason, the heat exchangers in the cases and coolers must be sized differently than those used for direct expansion operation. Additional loops can be employed if a large fraction of the refrigeration load is addressed by a secondary fluid temperature significantly different than 20 or –20°F. Possible examples of alternate loop temperatures are values of –10, 0, or 15°F, depending upon the air temperature required by these refrigeration loads. The use of multiple secondary fluid loops with temperatures closely matching the case air temperature requirements can result in more energy-efficient operation of the secondary loop system.

Secondary fluid loop piping consists of large main pipes that contain all fluid flow to and from the chiller. The piping is branched to the display case lineups as needed. The piping for the main and branched section is typically sized to have a fluid velocity of 4 to 6 ft/sec. In each lineup, further branching occurs to provide flow to each display case coil. A temperature control valve is used in each case coil to monitor case air temperature and to regulate fluid flow to maintain proper air temperature. The branch piping is also equipped with balancing valves that are used to set the branch flow rate. This adjustment is normally done at installation only.

The piping for the secondary fluid loops can be steel, copper, or plastic. The recommended type of plastic piping is constructed of high-density polyethylene (3-2). Insulation is also required. For the medium temperature piping, the recommended insulation is closed-cell foam (3-2). The recommended thickness of the insulation is 1/2 to 3/4 in. for piping underground or in air-conditioned space, 1-1/2 in. for pipe in non-conditioned space. For the low temperature piping, the insulation should be either styrofoam or polyisocyanurate foam. The recommended insulation thickness for low temperature piping is 1 in. for piping in air-conditioned space and 1-1/2 to 3 in. in non-conditioned space.

Secondary fluid flow rates are fairly high, since it is desirable to limit the temperature change of the fluid to 7 to 10°F while refrigerating the display cases. The resulting total flow rate for each loop is as high as 300 to 500 gpm. Because of the high viscosity of the secondary fluid at refrigerating temperatures and the fact that the fluid is circulated continuously, the energy associated with pumping is substantial and is a major component of the overall energy consumption of the secondary loop system. The flow rate of secondary fluid needed for each display case will vary as the refrigeration load changes. The total flow of fluid is also affected, but pump power remains constant, since control of the loop flow is achieved by bypass of flow around the pump. The discharge and total flow seen by the pump remain constant with this arrangement.

The central chiller systems employ multiple compressors that are piped in parallel. The compressors are multiplexed and are on/off cycled in response to the value of the suction pressure of the chiller evaporator. The chiller system employs either reciprocating or screw compressors. Screw compressors provide large refrigeration capacities needed to provide the refrigeration load associated with each fluid loop. Screw compressors do not exhibit as high an energy efficiency as reciprocating units, but are less susceptible to liquid refrigerant damage and, in general, are considered to be less of a maintenance issue. Reciprocating compressors can be used in the chiller system for better energy use characteristics.

Because of the location of the evaporator on the chiller skid, the compressors for the secondary loop system are considered close-coupled to the evaporator. The pressure drop and return gas heat gain are minimized in this configuration. Both these factors help to reduce compressor energy consumption.

The chiller system for the low temperature refrigeration can be equipped with mechanical subcooling. Two possible configurations can be used to supply the subcooling. The first consists of employing the medium temperature chiller to supply subcooling to the low temperature system. The piping arrangement for this type of mechanical subcooling is the same as that described for the multiplex system. A direct expansion heat exchanger is used to cool the liquid refrigerant used by the low temperature chiller system. The second approach is the use of an economizer heat exchanger with the screw compressors. The heat exchanger uses a portion of the liquid flow to refrigerate and subcool the remaining liquid. The vapor from the heat exchanger is piped to an intermediate pressure suction port on the screw compressor.

Secondary loop refrigeration systems employ hot secondary fluid to provide defrost to the display cases. In this system, a heat exchanger at the chiller is used to heat a portion of the fluid with the discharge gas from the compressors. The hot fluid is piped to the cases using a third pipe. The fluid leaving the defrosted case is returned to the chiller through the common return piping system. The use of hot fluid for defrost has been found to shorten defrost time substantially (3-3) and also reduce the amount of refrigeration associated with pull-down and recovery of the case to operating temperature. It is estimated that the energy use for defrost is about half of what is seen in multiplex systems employing hot gas defrost (3-3).

3.3.1 Secondary Fluids for Supermarket Refrigeration

The most commonly used secondary fluid in both commercial and industrial refrigeration applications is either ethylene or propylene glycol and water. Glycols are preferred because they are inert to all common piping materials and most nonmetallic gaskets and seals. Propylene glycol is also nontoxic and nonflammable and is the most suitable glycol for use in unattended operation. A propylene glycol – water mixture show reasonable heat transfer and pumping properties for medium temperature (+20°F fluid temperature), but have very high viscosity at glycol concentrations needed for low temperature refrigeration.

Commercial products exist for use as low temperature fluids and several investigations have been conducted (3-4,3-5) to identify possible candidate fluids for secondary loop systems. The desired properties of the candidate secondary fluids are that they are noncorrosive to common refrigeration construction materials, nonflammable under normal operating conditions, and are reasonably nontoxic. In addition to these requirements, good transport properties are needed to reduce pumping power. Heat transfer properties are of some concern, but of secondary importance compared to pumping.

The following is a listing of candidate fluids for use in secondary loop systems that have been cited in the literature (3-4,3-5). This list gives the general chemical name of each fluid along with several trade names.

- Propylene glycol/water.
- Potassium Formate/water.
  - Pekasol 50.
  - Freezium.
  - Hycool.
- Inhibited alkali ethanate solution.
  - Tyfoxit.
- Hydrofluoroether.
  - HFE-L-13938.
- Cyclohexene.
  - D-Limonene.
- Polydimethilsiloxane (Silicon Oil).
  - Syltherm.
  - Dowtherm.
- Synthetic Isoparaffinic Petroleum Hydrocarbons.
  - Therminol.

Evaluation of these fluids for use in secondary refrigeration was carried out by ref. (3-5) where the fluids were compared in terms of relative pumping power to the fluid Tyfoxit. The relation takes into account the flow rate of secondary fluid needed to address a refrigeration load Q, which is found from

$$Q = \dot{m}\ c\ \Delta T$$

where

$\dot{m}$ = the mass flow rate of the fluid
c = the specific heat of the fluid
ΔT = the temperature change of the fluid

The velocity of the fluid flow, V, can be determined from

$$\dot{m} = \rho\, V\, A = \rho\, V \frac{\pi D^2}{4}$$

$$V = \frac{4Q}{c \Delta T \pi D^2}$$

where

D = the diameter of the brine piping

The pumping power is found from the relation

$$\text{Pumping Power} = \frac{\dot{m}}{\rho}\,\Delta P$$

$$= \frac{\pi D^2}{4}\,V\,\Delta P$$

where

ΔP = the pressure drop across the fluid loop

The pressure drop is determined from the relation

$$\Delta P = f\,\frac{L}{D}\,\frac{\rho V^2}{2}$$

where

f = the friction factor
L = the total length of pipe

The friction factor, f, is calculated from

$$f = \frac{0.29}{Re^{0.2}}$$

where

Re = the Reynold number of the flowing brine

$$= \frac{\rho V D}{\mu}$$

where

μ = the kinematic viscosity of the brine

The relative pumping power is found by holding the refrigeration load, fluid temperature difference, and pipe diameter and length constant. The above relations are then combined to give an expression for pumping power in terms of fluid properties only relative to a base fluid

$$\frac{\text{Pumping Power}_A}{\text{Pumping Power}_B} = \left(\frac{\rho_A}{\rho_B}\right)^{-1.8}\left(\frac{\mu_A}{\mu_B}\right)^{0.2}\left(\frac{c_A}{c_B}\right)^{-2.8}$$

Figure 3-6 shows the relative pumping power for a number of candidate fluids, where Tyfoxit 1.20 is used as the baseline fluid. The figure shows only those fluids that had a relative pumping power close to that of Tyfoxit. Several fluids showed pumping power requirements that were 1 to 2 orders of magnitude greater. These fluids were Dowtherm, Syltherm, and HFE-L-13938. The fluid showing the lowest pumping power requirement was Pekasol 50 at a concentration of 90 percent.

3.4 Advanced Self-Contained Systems

A self-contained refrigeration system consists of a display case that has its own condensing unit mounted within the display case. Self-contained systems are presently used in supermarkets for a limited number of cases, where the cases are in a location inaccessible to refrigeration piping. An example is a refrigerated beverage case placed at the cash registers for spot sales. Self-contained units are also employed as add-on cases or for temporary display of special sales items.

Present self-contained display cases use small reciprocating compressors and air-cooled condensers. Heat is rejected directly into the sales area. Only a limited number of self-contained units of this type can be employed before noise and heat rejection levels interfere with store operation. Problems of this type caused store designers in the past to go to the remote machine room approach now used in most supermarkets.

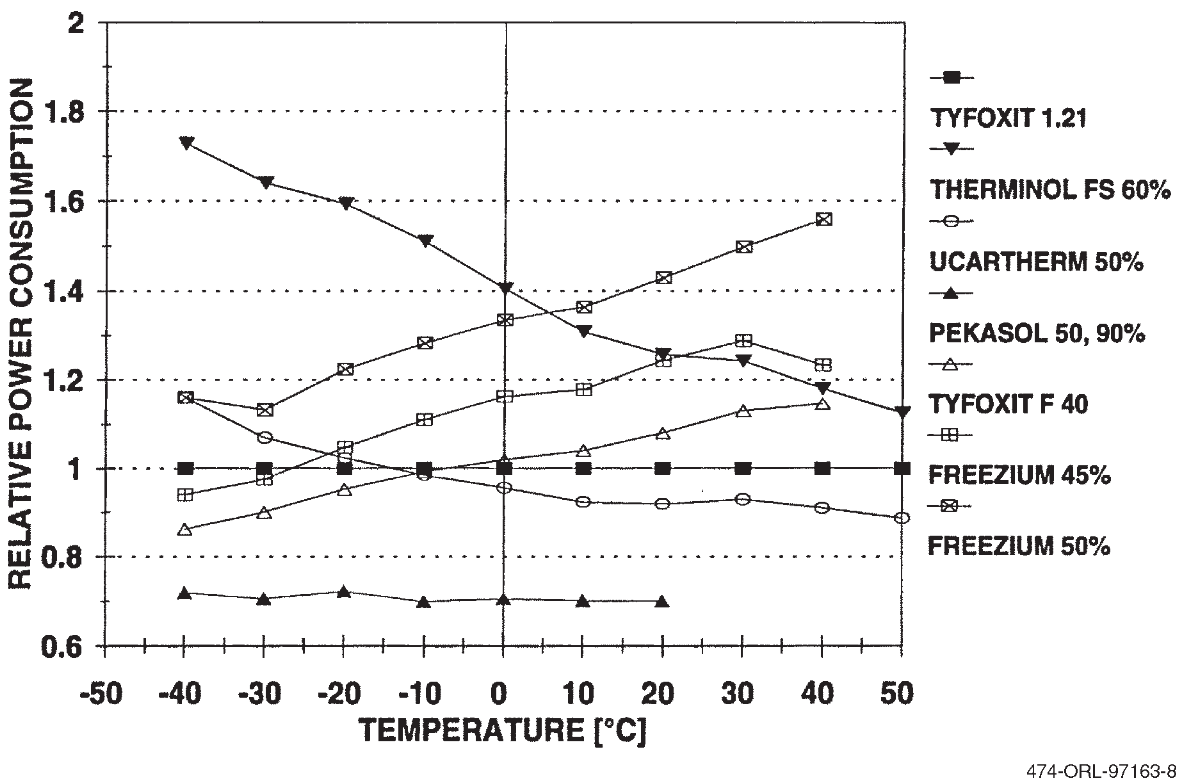

Figure 3-6. Pump power comparison of heat fluids relative to Tyfoxit 1.21 (3-5)

The self-contained system approach could be attractive for reduction of refrigerant charge. It has been estimated (3-6) that the total charge for a supermarket could be reduced to 100 to 300 lb of refrigerant if self-contained systems were used for all display cases.

An advanced self-contained system could be formulated, which used water-cooled condensers and a fluid loop for heat rejection. The fluid loop would be similar to that employed with the distributed system. The use of the fluid loop would eliminate concerns of heat rejection in the store sales area.

The compressor noise issue is still a factor that can be addressed by the use of scroll compressors. Until recently, scroll compressors were available only in a vertical configuration, which was not suitable for placement in display cases. Now, horizontal scroll compressors have been introduced, which could be employed for this purpose. These horizontal scrolls are capable of continuous unloading for capacity control and maintenance of a suction pressure set point. These scroll compressors are capable of unloading to as low as 25 percent of their full load capacity (3-7).

Figure 3-7 shows the relation between capacity control and compressor power required. The use of compressor unloading also allows the condensing temperature to vary as lower temperature heat rejection is possible with lower ambient temperature. For analysis, the minimum condensing temperature was set at 40 and 60°F for low and medium temperature refrigeration, respectively. This may or may not be practical, since 2 glycol loops are needed in order to have two different minimum condensing temperature values.

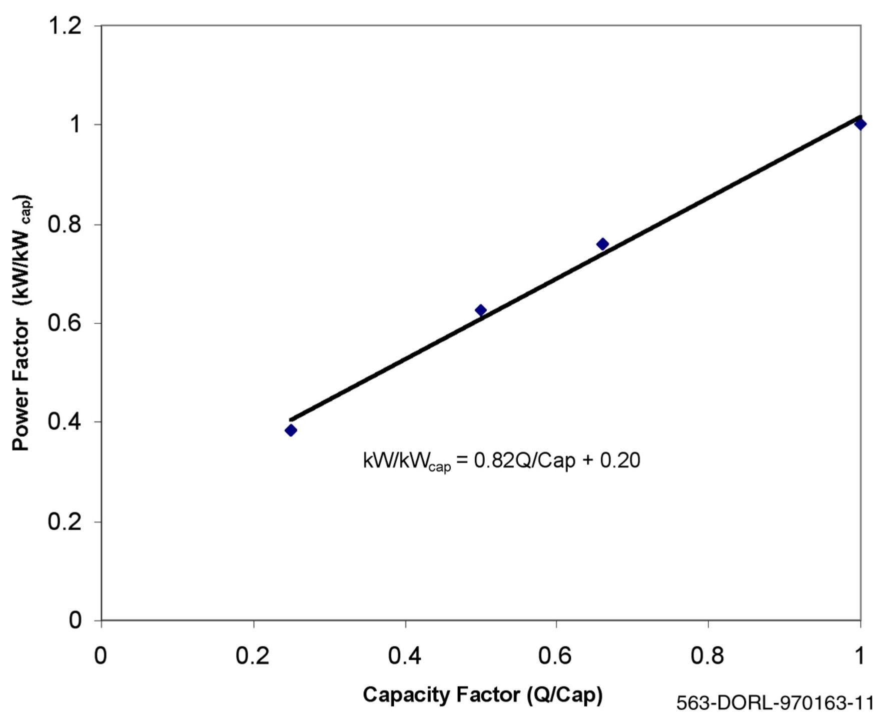

Figure 3-7. Relation between capacity and power used for modeling unloading scroll compressors

The close coupling of the compressor to the case evaporator seen in self-contained systems reduces the pressure drop at the compressor suction and also minimizes the heat gain to the suction gas. Both of these effects will result in more efficient operation and were included in the analysis.

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4. SUPERMARKET HVAC

4.1 HVAC Load Characteristics

Supermarkets have unique HVAC characteristics because of the large amount of refrigerated fixtures operated within the store. The refrigerated display cases remove a large quantity of heat and moisture from the store, which has several impacts on the HVAC requirements.

For space cooling, the cooling loads tend to have a small sensible portion and a large dehumidification need. The display cases are designed to operate at conditions of a 75°F dry-bulb and 55% RH, which is a lower temperature and humidity than normally seen in a commercial building. Humidity in the sales area will tend to increase the refrigeration load on the display cases, and will cause frost formation on the evaporator coils. Defrost of the coils is necessary in order to maintain the capability of providing refrigeration. Defrost increases energy consumption due to the use of electric heaters for defrost, if electric defrost is employed, and also increased compressor run time to lower the display case temperature after defrost. Frozen food door cases are equipped with anti-sweat heaters to prevent fogging of the glass doors. The amount of on time needed by these heaters is directly related to the store humidity level.

Store space cooling systems are often controlled using both a thermostat and humidistat. With this arrangement, the cooling system can be operated for dehumidification when the sensible cooling load has been satisfied. The resulting supply air is colder than desired. Either the store is allowed to cool to a temperature below the set point value, or reheat of the supply air is done to maintain the correct sales area temperature.

Space heating requirements tend to be greater for supermarkets because of the large amount of installed refrigeration. Figure 4-1 compares the heating requirement of a supermarket to that of other commercial buildings. The sensible cooling provided by the refrigeration tends to negate heat gains from store lighting and other electric uses. The heating season for supermarkets is longer than is seen in most commercial buildings for this reason.

To estimate HVAC loads in supermarkets, a so-called "case credit" is calculated to determine the effect of the refrigerated cases. The portion of the refrigeration load that impacts the store consists of the sensible and latent loads removed from the sales area, which is roughly 80 percent of the total case refrigeration load. The remainder of the case refrigeration load is associated with the removal of heat due to the operation of case fans, lights, and anti-sweat heaters. The latent portion of the load falls between 12 and 19 percent of the total case load, depending upon the type of case and evaporator temperature. The highest latent loads occur with either low or

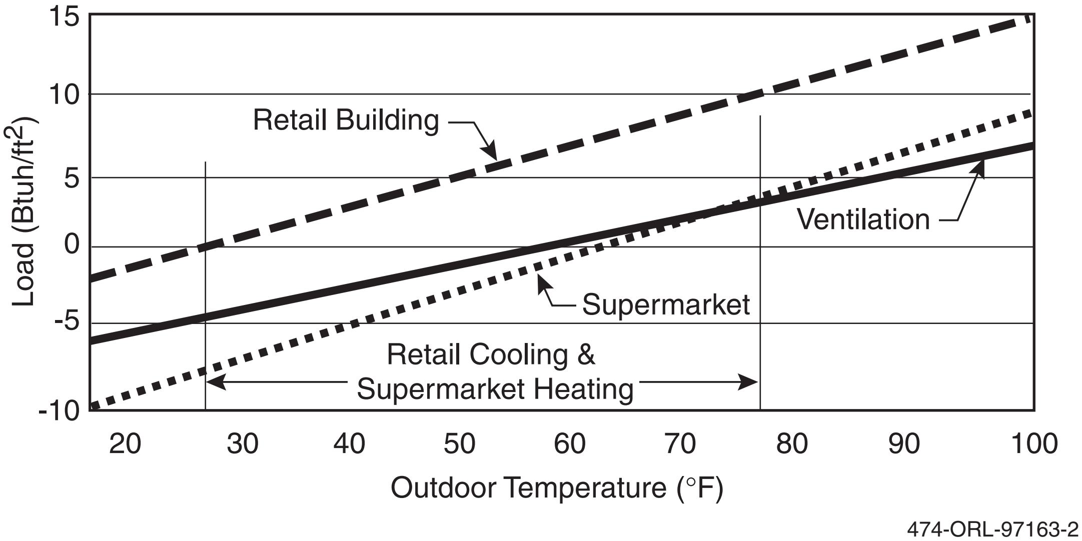

Figure 4-1. HVAC load characteristics for supermarkets and other retail buildings (4-1)

medium temperature multi-deck cases. Latent loads are reduced significantly when door or tub-type cases are employed.

Airflow rate for store air circulation is typically set at 1 cfm/ft^2 of sales floor area. Ventilation flow is set at 10 percent of total circulation, which corresponds to an air change rate of 4.5 per hour. Some stores are investigating the use of higher ventilation rates for improved indoor air quality. To meet the ASHRAE Standard 62-89, the ventilation flow rate to the store must increase to 30 percent of total circulation. This increase in ventilation flow has a significant impact on store HVAC requirements.

Supermarket air systems attempt to maintain the store under a positive pressure to eliminate infiltration. No exhaust flow is used in the HVAC other than that associated with cooking hoods, etc.

The refrigeration system is also a potential source of heat for space heating. Heat reclaim systems have been used in many stores where a portion of the reject heat from the refrigeration system is sent to a coil in the air ducting and is used to heat store supply air.

4.2 HVAC System Configurations

The layout of HVAC supermarket equipment falls into two configurations. The first consists of one or two air handlers located in the back of the store, with each air handler and its ducting operating as its own HVAC zone. The air handler (Figure 4-2) consists of a blower, and heating and cooling coils. Either one or two heating coils are employed with one being for refrigeration heat reclaim. The second coil consists of a duct heater that is either electric or gas fired. The cooling coil is placed upstream of the heating coils so that reheat of the supply air can be

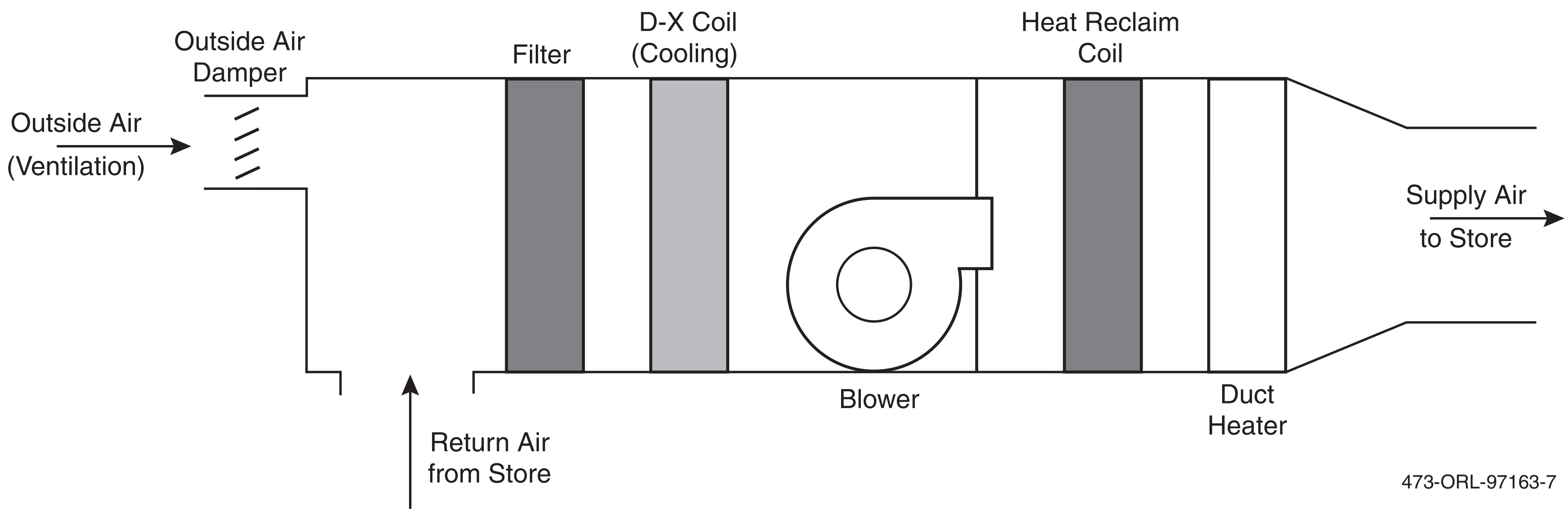

Figure 4-2. ***Elements of a supermarket air handler***

performed if desired. Ducting is provided to the air handler so that most of the airflow consists of recirculated air from the sales area. Ventilation air ducting is connected to the suction side of the air handler and a manual damper is employed to set the amount of ventilation air supplied. The air conditioning condensing unit for the cooling coil is located on the roof above the air handler. Supermarkets will have the air handlers mounted indoors in the duct systems with the heating and cooling equipment mounted on the roof above the air handlers. The alternate location for HVAC equipment is in a "penthouse" machine room located on the roof of the store. These penthouse systems are factory assembled and placed on the store roof where they are mated to the supply and return ducting. The penthouse will contain all elements of the HVAC system, including air handler elements and heating and cooling coils.

The second configuration, which has become more common, is the use of rooftop units (RTUs) at 4 to 8 locations on the roof above the sales area. Figure 4-3 shows a typical RTU used for commercial HVAC. Each RTU is equipped with an air handler consisting of a blower and ducting for supply, return, and ventilation airflows. The supply air duct contains heating and cooling coils. The RTU also has an air conditioning condensing unit with 2 to 4 compressors and an air-cooled condenser. Space heating is provided by a gas-fired duct burner.

4.3 Airflow Path Configurations

The most common airflow path used in supermarket HVAC is the single-path which is shown in Figure 4-4. This airflow configuration is used in most central and rooftop systems. The two airflows for return and ventilation air are joined at the suction side of the blower and then passed through the HVAC coils.

An alternate method now used in some installations is the dual-path approach, which is shown in Figure 4-5. Here the ventilation air stream is conditioned prior to mixing with the return air by a cooling coil located in the ventilation ducting. After mixing, the airflow is then directed through a second set of coils where final conditioning of the air is performed. The advantage of the dual-path method is that the latent load fraction associated with the ventilation air is often much higher than that of the return air. By dehumidifying the ventilation air and then mixing the amount of final conditioning needed is greatly reduced, particularly in terms of reheat. The use of dual path can eliminated the need for reheat in most applications. This method can be used with either vapor compression cooling or with desiccant dehumidification.

4.4 HVAC System Equipment

Standard supermarket HVAC systems consist of vapor compression air conditioners for space cooling, and gas duct heaters for space heating. These components are the same as those employed for most commercial and retail applications.

4.5 Desiccant Dehumidification

The use of desiccant dehumidification has been investigated for supermarkets. The advantage offered by desiccant systems is that they can address the latent load on the store

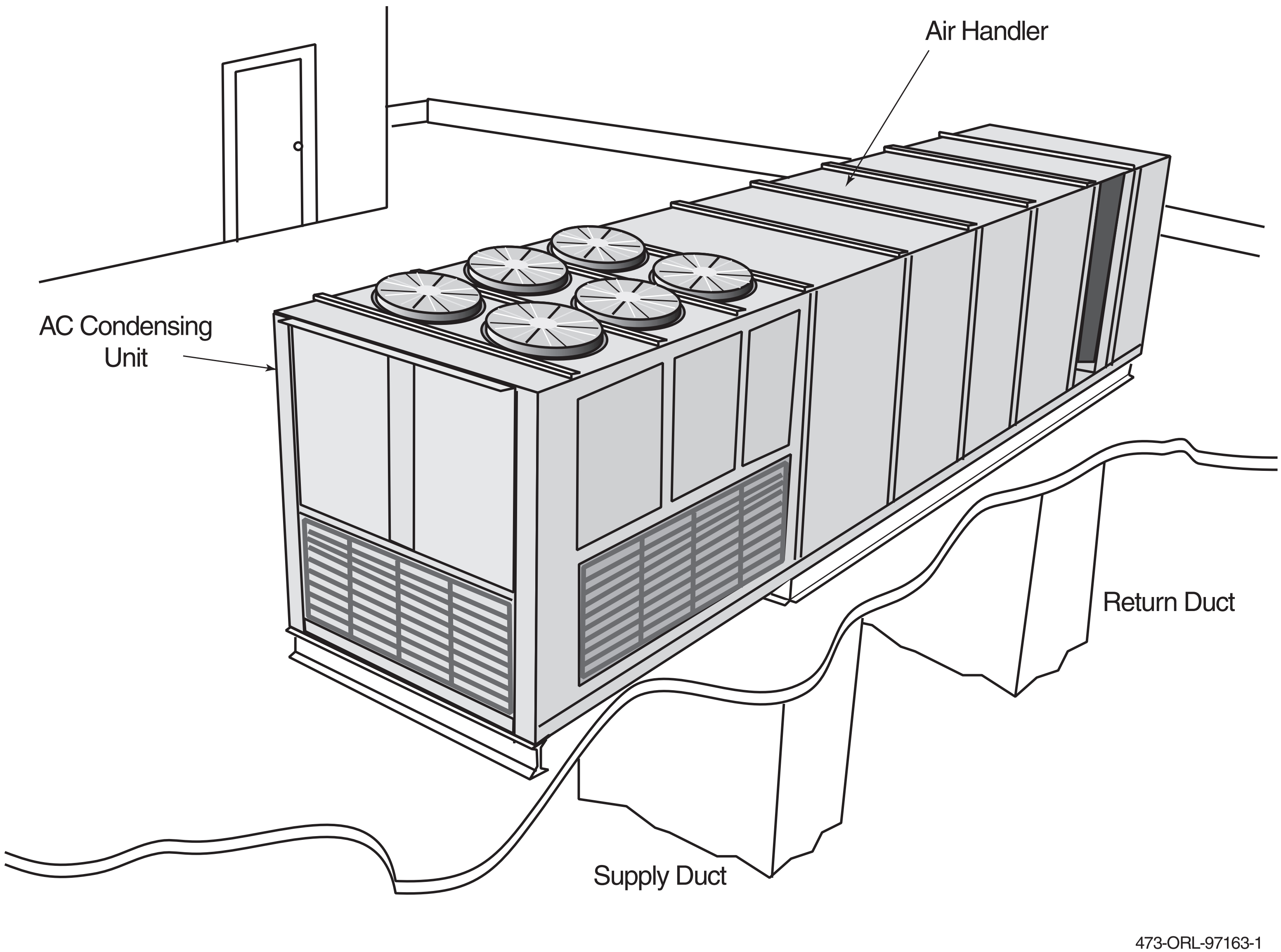

Figure 4-3. Rooftop unit for HVAC

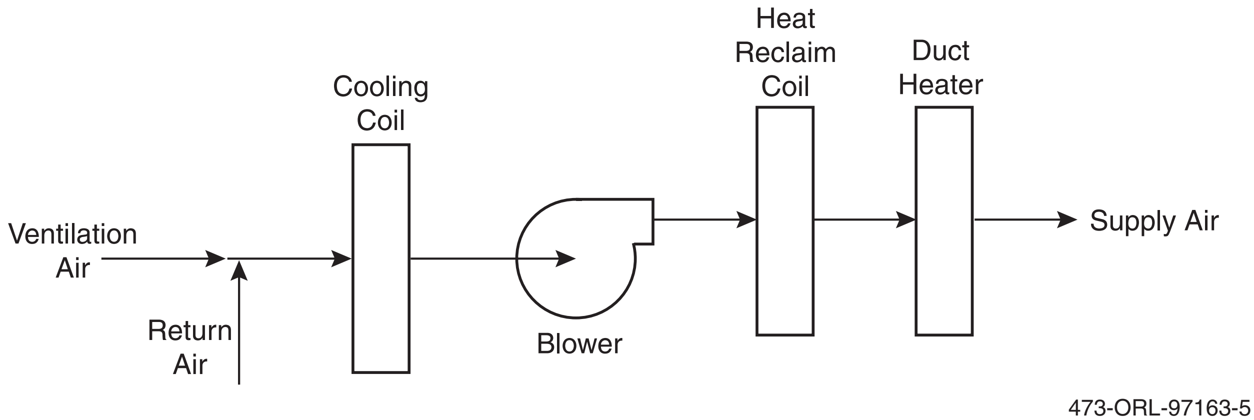

Figure 4-4. Single-path HVAC

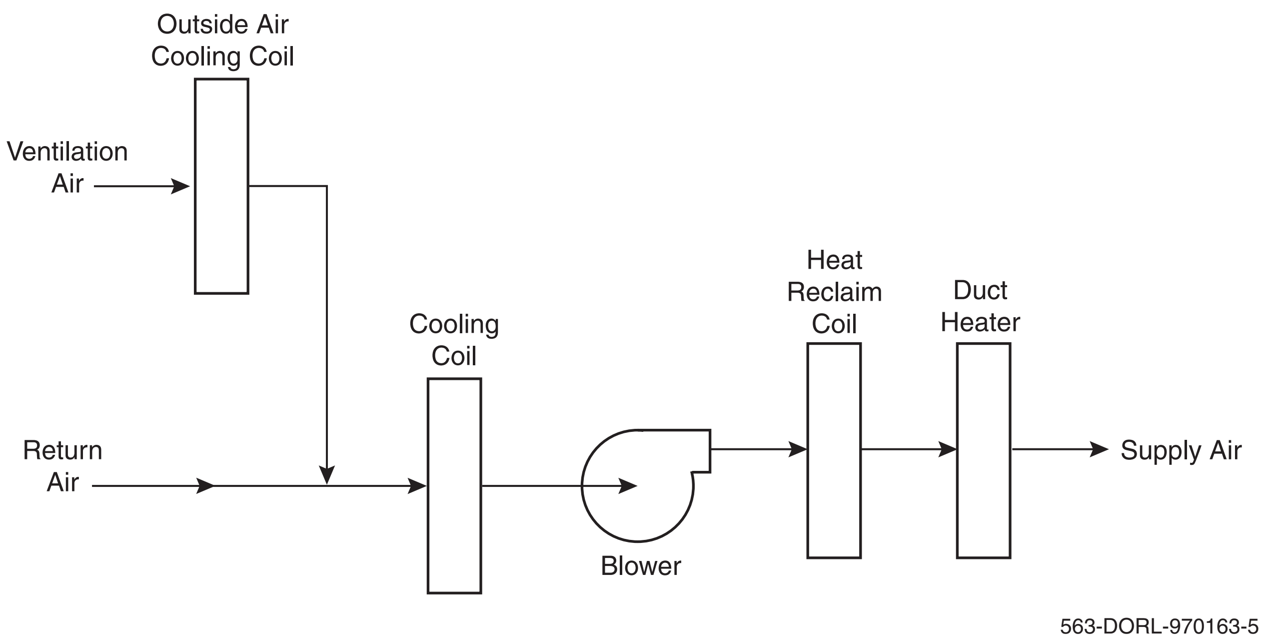

Figure 4-5. Dual-path HVAC

separately and convert the latent heat into a sensible cooling load that can be addressed as needed by a vapor compression cooling system.

The elements of a desiccant dehumidification system are shown is Figure 4-6. The air to be conditioned is passed through the desiccant wheel where moisture is adsorbed and the latent heat is released back to the air. Most desiccant systems also have a second wheel that is used as a rotary heat exchanger to cool the dried air and to preheat the regeneration air, which passes through the desiccant wheel in order to removes the moisture from the wheel. The regeneration air is passed through a gas-fired heater where it is heated to 180 to 220°F. Refrigeration heat

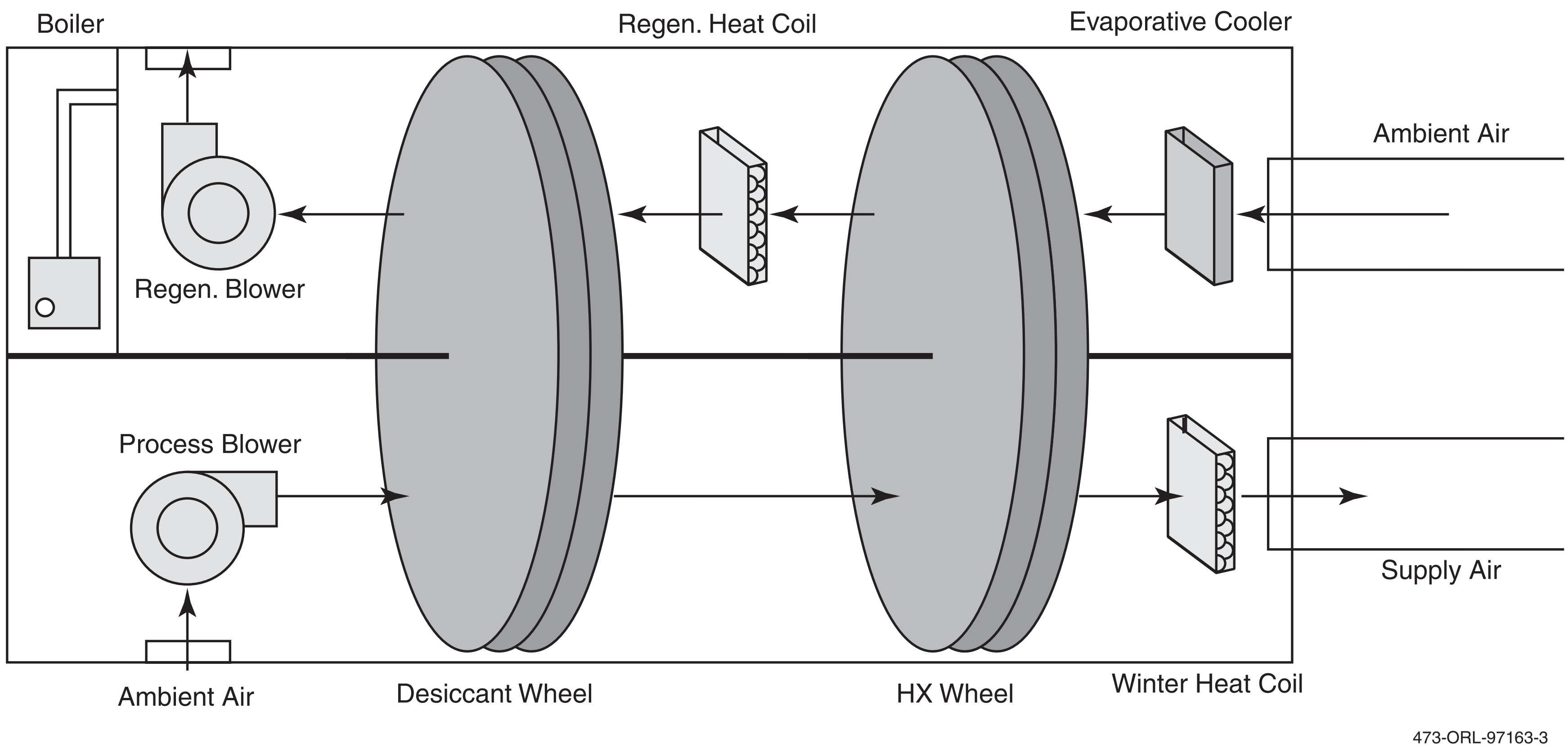

Figure 4-6. Elements of a desiccant dehumidification system

reclaim is employed on some systems to provide heat to the regeneration air, which helps reduce the amount of gas needed for operation.

Desiccant systems are used in a dual path arrangement where the desiccant system conditions outside air only. The air is dried thoroughly and is then mixed with the recirculation air. The air mixture is normally passed through a cooling coil to remove any excess sensible heat. One major advantage to this approach is that the need for reheat of the conditioned air is avoided.

A number of analyses have been carried out to evaluate desiccant systems in supermarkets (4-2 - 4-6). The results are extremely varied and totally dependent on local rates for electricity and natural gas. The general findings are that the amount of energy needed to dehumidify the entire store to a lower level is difficult to justify. While the refrigeration loads are reduced when humidity levels in the store are lowered, the reduction in refrigeration energy is not large enough to offset the costs associated with the added HVAC operation.

A case study done by the HEB supermarket chain compares desiccant and conventional HVAC systems in two operating supermarkets in the San Antonio, TX area (4-6). Table 4-1 gives a summary of the findings. In general, the desiccant system was more costly to install than the conventional system. Savings achieved by the desiccant system were negated by increased maintenance costs. Savings seen in refrigeration energy were on the order of 3 percent. The payback on the cost premium for the desiccant system was estimated to be 27.3 years.

The more accepted practice now used in a large number of supermarkets is to limit the use of desiccant systems (or any other low humidity system, such as dual path air conditioning) to areas in the store where lower humidity is very advantageous. The best example is the frozen food aisle of the store. The large amount of sensible cooling performed by the frozen food cases tends to cool the aisle too much, making it uncomfortable for customers. The desiccant system works well in this application, because the air from the desiccant system can be discharged directly into

Table 4-1. Comparison of first and operating costs for conventional and desiccant HVAC systems (4-6)

	Conventional Packaged Rooftop Unit	Desiccant
Store Conditions	75°F, 45% RH	75°F, 40%RH
Store Airflow (cfm/ft^2)	0.65	0.65
HVAC System First Cost ($)		
Components	48,000	107,000
Installation	12,400	15,640
Duct System (installed)	112,261	116,741
Total Installed Cost	172,661	239,381
Annual Expenses ($)		
Utility Cost		
HVAC	33,880	30,210
Refrigeration	55,366	53,740
Maintenance	1,041	3,895
Total Annual Expenses	90,287	87,845

the frozen food aisle, which helps offset the overcooling done by the cases. The drying of the air is limited to the vicinity of the low temperature refrigerated cases, where maximum benefit to the refrigerated cases is obtained. The frozen food cases are very susceptible to humidity in terms of frost deposition on the cases and product. The door case anti-sweat heaters are normally operated in response to the humidity level in the aisle and the drier air helps to limit heater operation.

4.6 Water-Source Heat Pumps

A water-source heat pump employs a vapor compression cycle to provide either space heating or cooling. Figure 4-7 shows a diagram of the operation of the heat pump. A water loop is used as the heat source for space heating and the heat sink for cooling. The vapor compression cycle is equipped with a valve that controls the flow of refrigerant depending upon the heat pump's mode of operation. For space heating, the water coil is used as the evaporator of the heat pump to remove heat energy from the water loop. The refrigerant is then pumped by the compressor to a higher temperature and sent to the indoor air coil, which acts as the refrigerant condenser. The condensing of the refrigerant provides air heating which is used for space heating. For space cooling operation, the flow of refrigerant is reversed and the indoor air coil serves as the heat pump evaporator. The air is cooled and dehumidified as it passes through the air coil by evaporating the refrigerant. The compressor moves the refrigerant from the evaporator and increases its temperature and pressure. The high-pressure refrigerant flows to the water coil, where the refrigerant is condensed and heat is rejected to the water loop.

The heating performance of the water-source heat pump is characterized by its heating coefficient of performance (COP_H) which is defined as

$$COP_H = \frac{\text{Heat Provided}}{\text{Compressor Work}} = \frac{Q + W}{W}$$

where

Q = the heat absorbed from the water loop
Tw = the work of compression

Values of COP_H for water-source heat pumps suitable for supermarket application fall in the range of 4.5 to 5.0 at an inlet water temperature of 70°F

Space cooling performance is specified by an energy efficiency ratio (EER) which is the ratio of the amount of cooling provided to the compressor power. The rated value of EER for water-source heat pumps is 10.0 Btu/hr/Watt at an entering water temperature of 85°F.

Water-source heat pumps are presently constructed in either single- or dual-path configurations. For the dual-path units, separate heat pumps are employed for the outside and circulated air flows. Each heat pump has dedicated compressors, and water and air coils. A

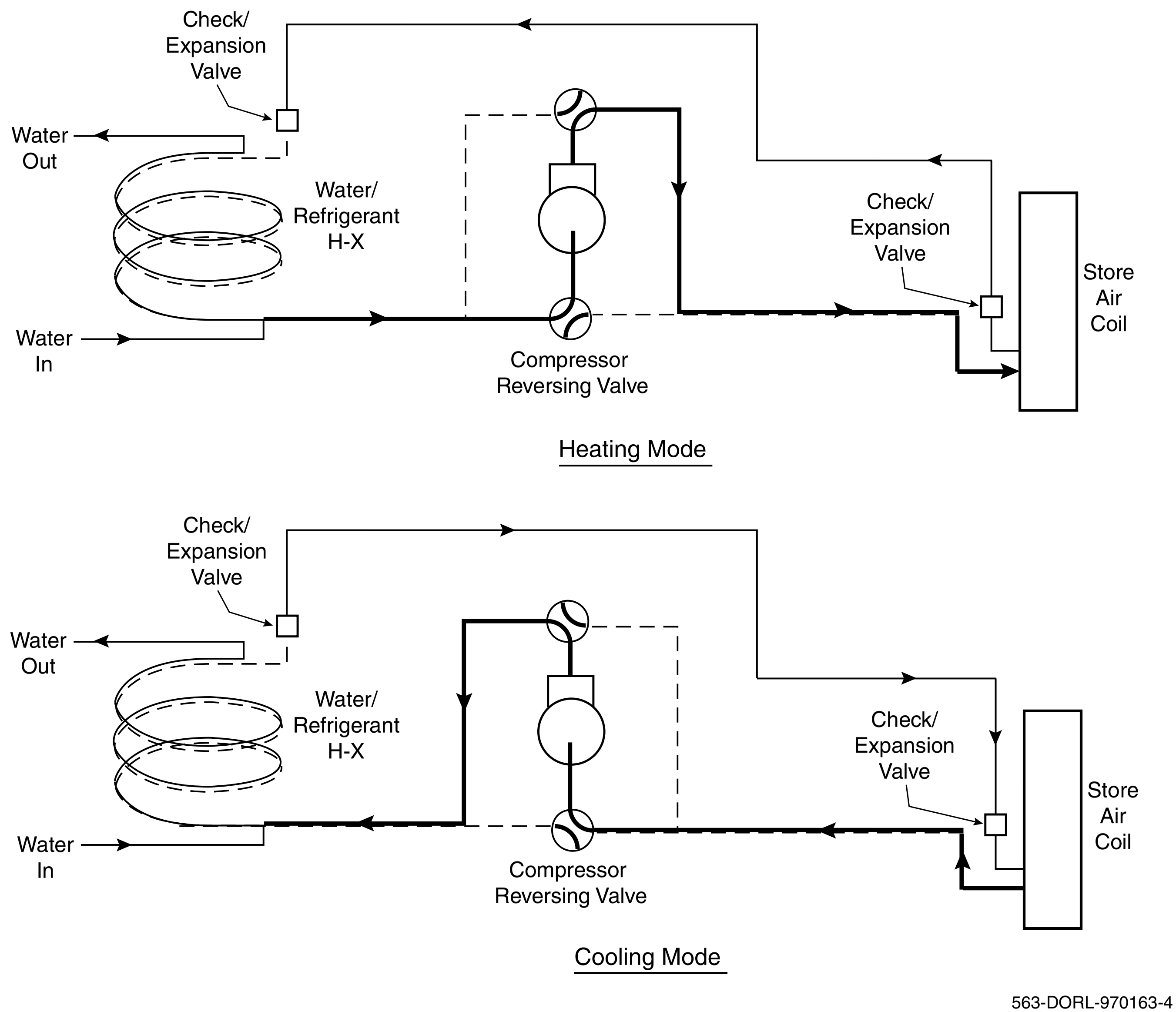

Figure 4-7. Detailed diagram of the water-source heat pump

single fan is used for air circulation and ventilation flows. The flow rate of ventilation air is controlled by a damper.

Water-source heat pumps can be used with supermarket refrigeration systems that employ a fluid loop for heat rejection. The integration of these two systems is shown in Figure 4-8. Figure 4-9 shows a detailed diagram of the water-source heat pump. The water-source heat pump is operated from a separate parallel branch of the fluid loop. In this arrangement, the fluid loop can be used to either provide heat to the heat pump during space heating, or provide heat rejection during space cooling. During heating operation, a control valve is used as a bypass to the fluid cooler to maintain a minimum temperature in the fluid loop.

Section 4 References

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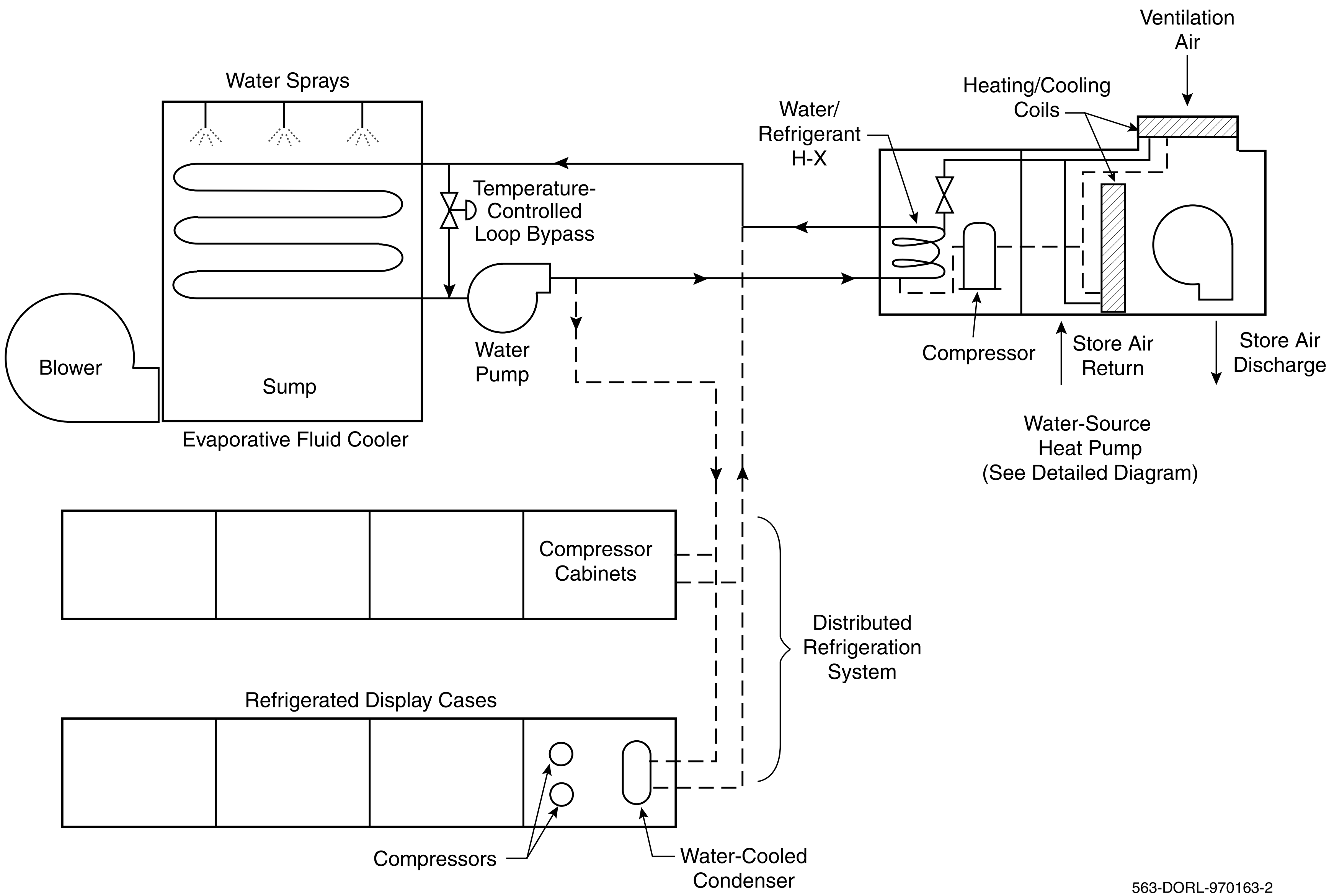

Figure 4-8. Integration of supermarket refrigeration and HVAC systems

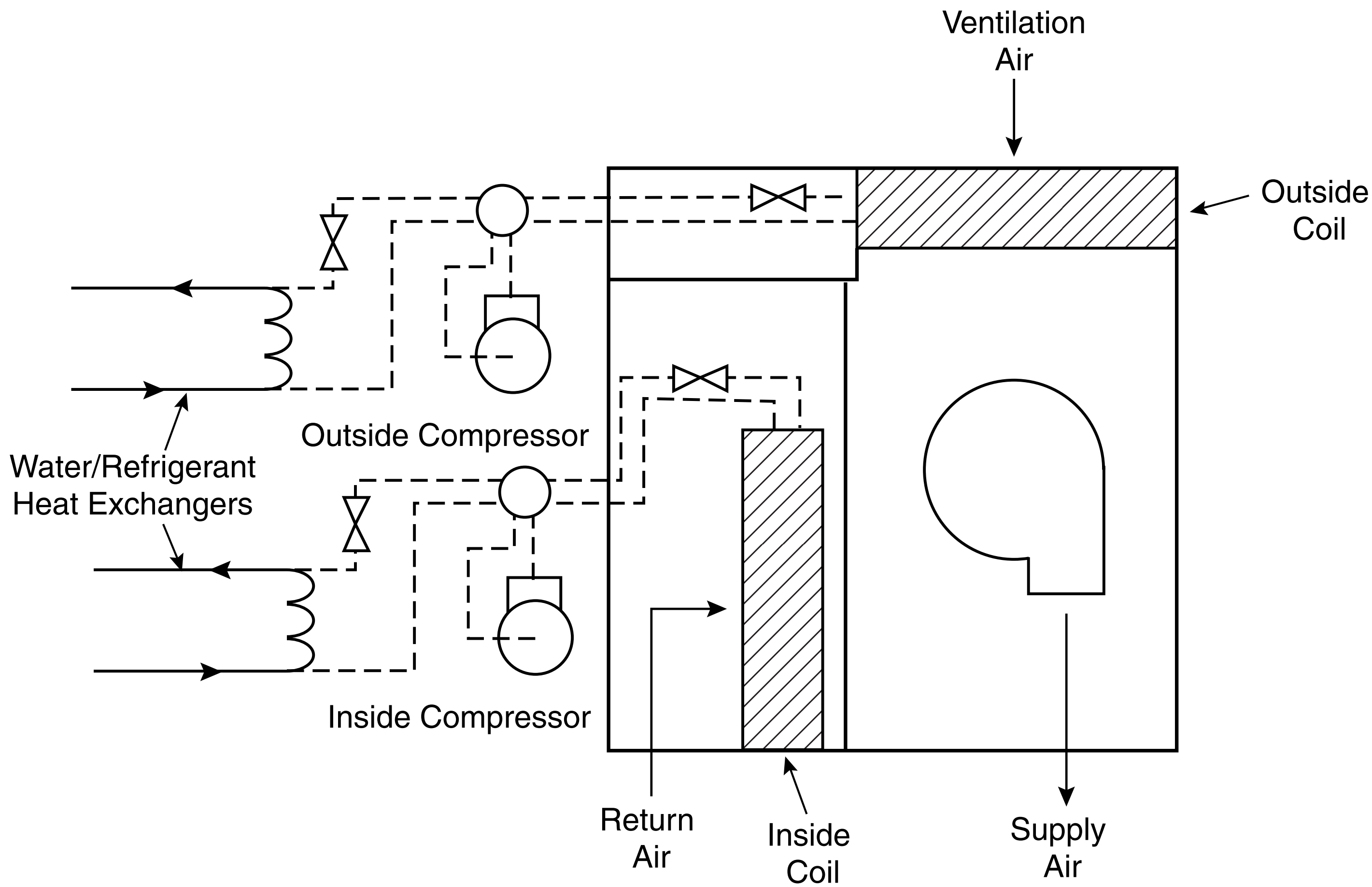

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Figure 4-9. ***Detailed diagram of water-source heat pump***

5. ANALYSIS OF SUPERMARKET REFRIGERATION AND HVAC

5.1 Multiplex Refrigeration

Figure 5-1 shows the flow chart of the model employed to analyze refrigeration system performance. Some variations exist in the model, depending upon the type of system examined (i.e., multiplex, distributed, secondary loop, etc.), but the overall procedure and methodology is the same for all. The annual performance is calculated on the basis of ambient dry-bulb temperature bins, where each bin specifies: an ambient dry-bulb temperature value; the coincident value of the wet-bulb temperature; and the number of hours at which the ambient temperature occurs during the year. The general procedure is, for each temperature bin, to calculate the power needed by the refrigeration system and apply that power to the number of hours at each ambient temperature. The procedure is repeated for all ambient temperatures seen at the site.

The bins can be constructed in any temperature increment desired, depending upon the weather data available. For all results presented here, a temperature bin size of 5°F, was employed, and the temperature values and hours observed were obtained from ref. (5-1).

A refrigeration configuration must also be specified, where each refrigeration system employed in the supermarket is described in detail. Table 5-1 lists the information that must be called out for the configuration specification of each refrigeration system. System information identifies the type of system, operating temperature level - low or medium, the design refrigeration load, lowest display case evaporator temperature in the system, minimum condenser temperature, type of condenser (and heat rejection for fluid loops), and the refrigerant employed. Optional information that can be specified include saturated temperature change between the display cases and the suction of the compressors, which is representative of the suction pressure drop, and the temperature rise in the return gas.

The first step in the analysis is to determine the refrigeration load on the system. Past experience (5-2) has shown that the refrigeration load will vary with outside ambient temperature, decreasing as the ambient temperature decreases. The rate of decrease in load is greater for medium temperature than low temperature refrigeration, and a minimum is reached, which is due to the space heating of the store. Store temperature will not fall below 68°F. The minimum store temperature is normally seen at ambient temperatures that are less than 40°F. The maximum values of store temperature and humidity are also constrained by the use of air conditioning, which tends to maintain the temperature at 75°F with a corresponding relative humidity of 55 percent. Maintaining the indoor conditions at these levels can be expected at outdoor ambient temperatures of greater than 85°F. Based on these constraints, a load factor can

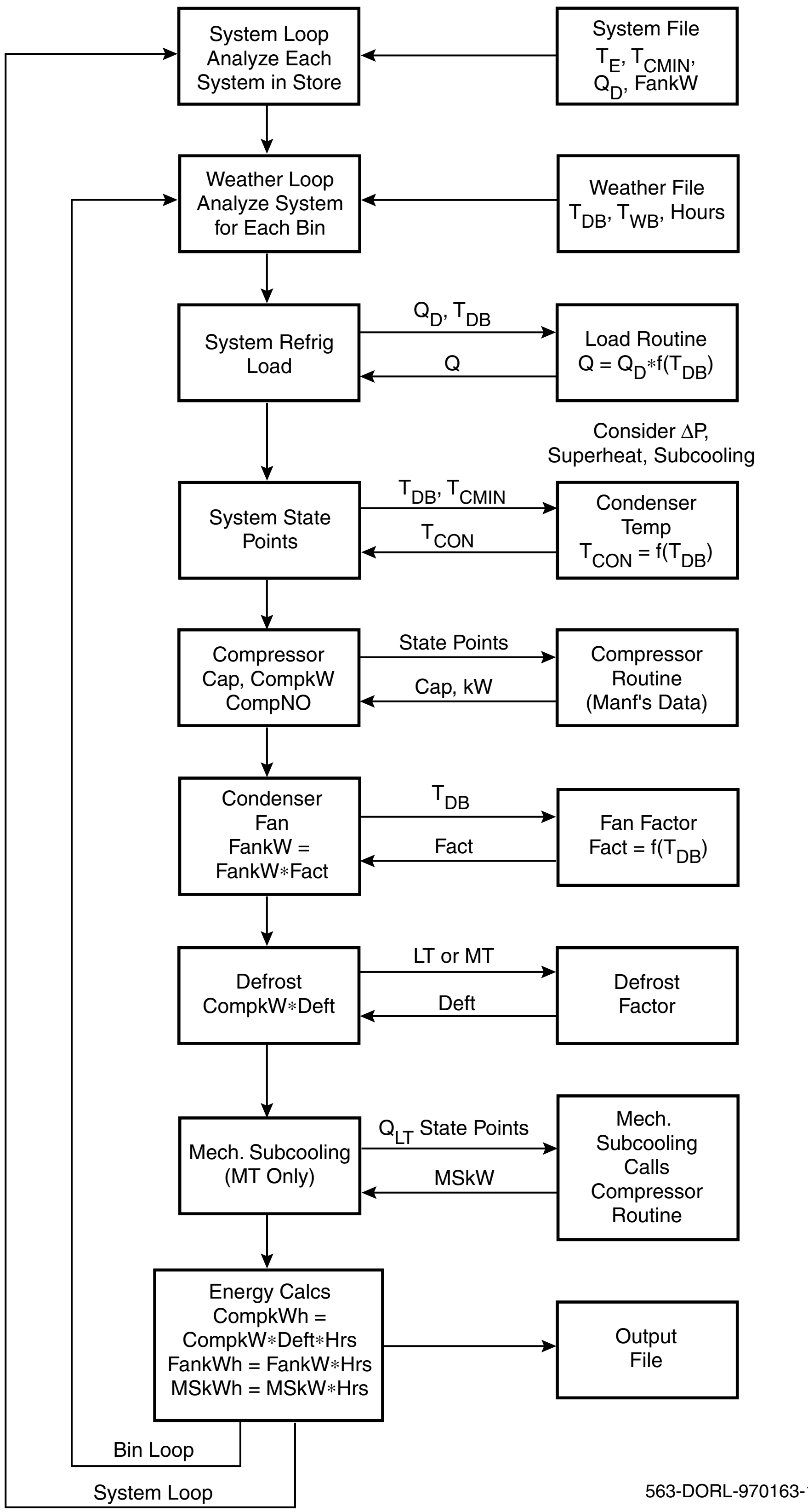

Figure 5-1. Supermarket refrigeration analysis model

Table 5-1. Refrigeration system characteristics needed for analysis model input

Specified Item	Description	Range of Values
Type of System	Type of refrigeration system being examined	Multiplex (incl. Low-Charge) Distributed Secondary Loop Adv. Self-Contained
Operating Temperature Level	Either low or medium temperature	LT – evap temps of –30 to -10°F MT – evap temps of 10 to 35°F
Subcooling	Denotes if mechanical subcooling is employed with this system	SUB – subcooled NS – no subcooling
Refrigeration Load (Q_D)	Design refrigeration load for this particular system	Total of refrigeration loads specified by the case and walk-in cooler manufacturers
Evaporator Temperature (T_E)	Lowest case or cooler evap. temperature associated with the system	
Minimum Condenser Temperature (T_{CMIN})	Lowest condensing temperature employed with the system	Dependent on compressor type Reciprocating Screw Scroll
Condenser Type	Type of condenser employed	AIR – air-cooled EVAP – evaporative WATER – water-cooled
Fluid Cooler Type (Water-Cooled Condensing Only)	Type of fluid cooler employed with water-cooled condensers	DRY – air-cooled WET – evaporative
Refrigerant	Refrigerant employed in system	22,134a,502,404A,507, 407A, 407B
Optional Inputs		
Saturated Suction Temperature Drop	Change in saturated temperature between the case evaporator and the compressor suction (i.e., suction pressure drop)	Specified in °F
Return Gas Temperature Rise	Increase in return gas temperature	Specified in °F

be calculated which is applied to the design load to determine the refrigeration load for each temperature bin. The refrigeration load factor is determined from the following relations:

- For both low and medium temperature refrigeration, the value of the refrigeration load is set at design for ambient temperatures of 85°F or greater.

- At ambient temperatures of 40°F or less, the refrigeration load is at its smallest fraction of design value, which is 66 percent for medium temperature and 80 percent for low temperature.

- For ambient temperatures between 40 and 85°F, the load factor is found from

$$\text{Load factor} = \left(1 - (1 - \text{min})\frac{(85 - T_{amb})}{(85 - 40)}\right)$$

where

min = the minimum fraction of design load (0.66 for medium temperature and 0.8 for low temperature)

T_{amb} = ambient dry-bulb temperature

The state points for the refrigeration system are then determined. The system configuration specifies the desired evaporator temperature for the display cases. In operation, pressure drop will occur between the case evaporators and the compressor suction. This drop is reflected in a lower saturated temperature value at the compressor suction. Heat gain to the return gas will also take place, which affects the mass flow rate of refrigerant seen by the compressor. Both of these factors tend to decrease the capacity of the compressors and increase the run time need to meet the refrigeration load. The amount of pressure drop and superheating is a strong function of the distance between the display cases and the compressors, increasing with increased distance. In the analysis, these factors are taken into account by values included with the system configuration description.

The condenser temperature is also determined at this time. The most significant parameter in determining condensing temperature is the ambient temperature, since heat is rejected by heat transfer to the ambient. The operation of the condenser can be characterized by the temperature difference (TD) between the condensing refrigerant and the ambient. The condensing temperature is allowed to vary with the ambient until a certain minimum condensing temperature is reached. At that point, control of the condenser fans or a liquid pressure regulator maintains the condensing temperature at the minimum value. The model compares the ambient temperature with the characteristic TD of the condenser type specified and calculates a condensing temperature. The calculated value is compared with the specified minimum condensing temperature. If the calculated temperature is less than the minimum, the minimum value is used to set the state point of the refrigeration system.

The condenser TD is determined by the type of condenser modeled. A summary of the characteristics of each condenser and heat rejection system examined here is listed in Table 5-2. For air-cooled condensers, the TD refers to the difference between the condensing and ambient dry-bulb temperatures. The standard values of TD for air-cooled condensers in supermarket refrigeration are 10 and 15°F for low and medium temperature, respectively. The TD of an evaporative condenser refers to the difference between condensing and ambient wet-bulb temperatures. Evaporative condensers are often sized to produce a condensing temperature of 100°F at the design wet-bulb, however, analysis and field measurements by Foster-Miller (5-3) showed that the lowest combined compressor and condenser fan power is achieved at a TD value

Table 5-2. Heat rejection system specifications for refrigeration modeling

System	Temperature Difference ($T_{cond} - T_{amb}$)	Design Load/kW Fan Power (kBtuh/kW)
Air-Cooled Condenser	TD = 10°F Low temperature TD = 15°F Medium temperature	34 Low temperature 65 Medium temperature
Evaporative Condenser	TD = 12°F based on wet-bulb temperature	70.5 Condenser fan and spray pumps
Water-Cooled Condenser Wet Fluid Cooler	TD = 10°F based on water temperature Tower approach = 12°F based on wet-bulb temperature	70.5 tower fan and spray pumps
Water-Cooled Condenser Dry Fluid Cooler	TD = 10°F based on water temperature Tower approach = 12°F based on dry-bulb temperature	65 tower fan only

of approximately 12°F. This lower value was used for all relevant analysis presented here. The TD of a water-cooled condenser is defined as condenser temperature – inlet water temperature, which is typically about 10°F. The inlet water temperature is affected by the ambient, and whether dry or evaporative heat rejection is employed at the fluid cooler. For dry rejection, the outlet water temperature is 10 to 15°F higher than the ambient dry-bulb temperature. For evaporative rejection the water temperature is about 10 to 15°F higher than the wet-bulb temperature. Water-cooled condensers are specified with either wet or dry heat rejection. The wet system will have a condensing temperature either the same or slightly less than that seen with air-cooled condensing. Use of the dry tower results in higher condenser temperatures, because of the added temperature difference seen across the water loop.

The refrigerant liquid temperature is also determined to set the state points of operation. For non-subcooled systems, the liquid temperature is taken at 10 deg less than the condensing temperature. When mechanical subcooling is employed in multiplex systems, the liquid refrigerant temperature leaving the subcooler heat exchanger is typically 40°F. Some warming of the liquid occurs as the liquid is piped to the display cases such that the temperature of the liquid entering the cases is set at 42°F. The refrigeration needed for mechanical subcooling is normally provided by the highest temperature refrigeration system in operation, typically at a SST of 25 to 35°F. The size of the mechanical subcooling load varies with the load of the subcooled system, normally the low temperature refrigeration. For each temperature interval the low temperature refrigeration load is first determined, and the liquid refrigerant flow required for this load is then determined. The subcooling load is the amount of cooling needed to lower the temperature of the liquid flow from 10°F less than the condensing temperature to 40°F. The subcooling load is added to the refrigeration load of the medium temperature system and is used to determine the energy consumption for that system.

Once the state points are determined, the capacity and power of the compressors is found. These calculations are made using the compressor performance data supplied by the manufacturers. Performance data consists of refrigeration capacities (Btuh), refrigerant mass flow rate (lb/hr), and input power (Watts) as functions of saturated suction temperature (SST) and saturated discharge temperature (SDT). These data are obtained from the ARI equation

$$\begin{aligned}\text{Capacity, Mass Flow, or Power} = {} & C_0 + C_1{*}SST + C_2{*}SDT + C_3{*}SST^2 \\ & + C_4{*}SST{*}SDT + C_5{*}SDT^2 + C_6{*}SST^3 \\ & + C_7{*}SDT{*}SST^2 + C_8{*}SST{*}SDT^2 \\ & + C_9{*}SDT^3\end{aligned}$$

where

$C_0 \ldots C_9$ = Performance equation coefficients
SST = Saturated suction temperature (°F)
SDT = Saturated discharge temperature (°F)

Compressor manufacturers provide three sets of coefficients for each compressor, where each set is to determine either cooling capacity, mass flow rate, or compressor input power.

The compressor cooling capacity and mass flow rate given by the above equations are determined at particular rating conditions. One such condition commonly seen is a return gas temperature of 65°F and 0°F of liquid subcooling. Corrections are made to account for the values of superheat and refrigerant liquid temperature. The superheat correction takes into account the density and enthalpy change, while change in liquid temperature affects the enthalpy of the refrigerant entering the evaporator. The correction applied to the compressor capacity or mass flow rate is

$$\textit{Capacity}\ \text{correction} = \frac{v_r}{v} * \frac{(h_{out} - h_{in})}{(h_{rout} - h_{rin})}$$

where

v = the specific volume of the refrigerant (ft^3/lb)
h_{in} = the enthalpy of the refrigerant entering the evaporator
h_{out} = the enthalpy of the refrigerant leaving the evaporator
the subscript, r, designates that the property is at the rating conditions

The capacity value and refrigeration load are then used to find the number of compressors operating by taking the ratio of refrigeration load to capacity. Typically 3 or 4 compressors are needed to meet the load at design conditions. At other conditions less than this number is required. Fractional values represent compressor on/off cycling. The analysis does not use specific compressor models, but instead uses a single generic size for each type of compressor. The generic size is based upon the most commonly compressor, which is a 7.5 HP unit for the reciprocating compressor and a 6 HP unit for a scroll compressor.

Compressor energy consumption for the temperature bin is found by first determining the power needed by the compressor at the state point and load conditions found. The compressor power is multiplied by the number of compressors operating and the number of hours associated with the ambient temperature.

The fan power for remote condensers or fluid coolers is based upon the type of condenser or cooler being examined. Table 5-2 gives the values of fan power required as a function of the design refrigeration load. Air-cooled condensers for low temperature refrigeration are sized for a smaller TD and require more fan power than condensers employed with medium temperature refrigeration. Fan requirements are less for evaporative heat rejection than is needed for dry rejection, because less air flow is needed. The power value listed in the table for the evaporative units includes the sump pump that is employed to spray water over the heat exchanger coil. Installed fan power is found by dividing the design refrigeration load by the appropriate value found in the table.

The condenser fans operate continuously as long as the resulting condensing temperature is greater than the specified minimum value. Fan cycling is employed with both the condensers and fluid coolers to regulate the condensing temperature when full operation of the fans reduces condensing temperature below the minimum value. Fan energy is estimated by multiplying the installed fan power by a fan factor, that represents the amount of fan on-time needed to maintain the condensing temperature at the minimum. For air-cooled condensers and dry heat rejection, the analysis sets the fan factor at 1.0 when the sum of the ambient dry-bulb temperature and the condenser TD are greater than the minimum condensing temperature. The fan factor is set at 0.25 when the ambient dry-bulb temperature is less than 30°F. For ambient temperatures greater than 30°F where continuous fan operation is not needed, the fan factor is calculated from

$$\text{Fan Factor} = \left(1 - 0.75 * \frac{(T_{con} - TD - T_{amb})}{(T_{con} - TD - 30)} \right)$$

where

T_{con} = the minimum condensing temperature (°F)
TD = the temperature difference (°F)
T_{amb} = the ambient dry-bulb temperature (°F)

For evaporative rejection, the above relation is also employed, but the ambient wet-bulb is used instead of the dry-bulb temperature. The fan energy for the temperature bin is determined from the product of the installed fan power, the fan factor, and the number of hours in the bin.

The systems employing fluid loops for heat rejection also require a circulation pump to circulate the water/glycol between the water-cooled condensers and the fluid cooler. For the analysis, the power requirement for this pump is 6 kW. Pump operation is continuous, so that the annual energy consumption for the pump is 52,560 kWh. This amount is added to the fan energy to determine the total energy consumption for heat rejection for the fluid loop heat rejection.

Defrost must also be accounted for in the energy calculation. Refrigeration systems considered here do not employ electric defrost. Hot gas and off-cycle defrost are used for low and medium temperature systems, respectively. Added compressor energy consumption is seen with either of these defrost methods, due to added cooling that must be provided to return the cases to operating temperature after defrost. From previous field tests conducted by Foster-Miller (5-4), the added energy seen is approximately 2 and 3 percent of compressor energy consumption for medium and low temperature refrigeration, respectively. These factors are applied to each temperature bin result.

The energy for mechanical subcooling of the low temperature refrigeration is addressed by the medium temperature system with the highest SST. As mentioned previously, the subcooling load is calculated for each temperature bin and is added to the refrigeration load of the appropriate medium temperature system. The compressor run time for the medium temperature system is calculated on the basis of this combined load.

The bin calculation is repeated until energy values are set for each temperature bin for the refrigeration system configuration specified. Once the bin loop is completed, the model then obtains the next system description and the bin loop is repeated for this next system. The procedure continues until all systems are analyzed.

5.1 Analysis Variations for Distributed and Secondary Loop Refrigeration

5.2.1 Modeling of the Distributed Refrigeration System

The modeling and analysis of the performance of the distributed refrigeration system follows the same procedure described above. The compressor cabinets contain multiple scroll compressors that are piped in parallel. These compressors are on/off cycled in the same fashion as the multiplex compressors to provide control of suction pressure. The same procedures to determine the refrigeration load and state points are followed. The energy consumption of the compressors is found by first comparing the refrigeration load to the available capacity to determine the number of compressors needed. The power requirement of each compressor is then determined from the state points and performance data. The energy is the product of the number of compressors, the power, and the number of hours in the temperature bin. The exceptions to the analysis procedure are for liquid refrigerant subcooling and heat rejection, using the glycol/water fluid loop.

The scroll compressors are capable of mid-scroll injection of refrigerant vapor, which can be used to subcool liquid refrigerant. The subcooling is done using a heat exchanger in the liquid line that is mounted in the compressor cabinet. A portion of the liquid is taken from the liquid line and is expanded into the exchanger to provide liquid subcooling. The vapor generated at the heat exchanger is piped to the scroll compressor at their injection port. The performance obtained by mid-scroll injection subcooling is determined from manufacturer's performance data for the scroll compressor when subcooling is applied. The resulting subcooled liquid temperature is taken at 50°F.

The heat rejection system for the distributed refrigeration system consists of water-cooled condensers mounted in each of the compressor cabinets. A glycol/water loop is pumped between the condensers and a fluid cooler. Either an evaporative or dry fluid cooler can be used in the analysis. The condensing temperature of the compressor cabinet is set at the inlet fluid temperature plus a temperature difference of 10°F. The water temperature is set by the type of fluid cooler employed. For evaporative units the water temperature is the ambient wet-bulb temperature plus 15°F. For the dry cooler, the water temperature is the ambient dry-bulb temperature plus 15°F. A minimum water temperature exists that produces the lowest allowable condensing temperature. The analysis does not allow water temperature to drop below this minimum. Fan and pump energy are determined using the same approach described above for heat rejection.

5.2.2 Modeling of Secondary Loop Refrigeration

The major difference in the analysis of the secondary loop refrigeration system is the operation of the secondary loop. The loop consists of brine that is pumped between a central chiller and the display cases. Several secondary loops are employed in a supermarket, depending upon the refrigeration loads required at each loop temperature. For the supermarket modeling, four loops were considered operating at brine temperatures of -20 for low temperature, and 10, 20 and 30°F for medium temperature refrigeration, respectively. The analysis is, therefore conducted separately for each of these loops at each ambient temperature. Energy results are combined at the completion of the analysis to determine total energy consumption.

The system configuration specifies the design refrigeration load for each secondary loop. The analysis first considers the variation on the refrigeration load seen at each ambient temperature, using the method described previously. The model assumes that the refrigeration load at the display cases is handled by a constant temperature change of the fluid, while the flow through the cases is varied as the refrigeration load varies. This flow arrangement is an attempt to simulate the operation of a temperature control valve that maintains constant fluid outlet temperature from the display case heat exchanger. Since all loads on the loop behave in this fashion, the estimated total fluid flow can be found from

$$\dot{M}_{brine} = \frac{\Delta T_{brine}}{C_{brine}}$$

where

$\dot{M}_{brine}$ = the mass flow rate of brine (lb/hr)

ΔT_{brine} = the temperature change in the brine seen at the cases

C_{brine} = the specific heat of the brine

The secondary fluid loop will experience some heat gain while flowing between the cases and the central chiller. The most significant of these gains is the addition of energy do to operation of the secondary fluid loop pump. The pump power is based on the maximum fluid flow needed to meet the design refrigeration load, which is found with the above relation. The required pump head is set at 75 ft of water column (WC) for low temperature refrigeration loop and 50 ft (WC) for the medium temperature loop. While the flow to the cases varies as the load changes, the total fluid flow through the pump remains constant, since a bypass is used to regulate the operation of the pump in the loop. The power input to the pump is calculated as the ideal power for the maximum fluid flow and head divided by a pump efficiency of 55 percent. The power input to the pump is converted into heat in the fluid, which must be removed by the chiller system. The rise in temperature is calculated from

$$\Delta T_{pump} = \frac{\text{Pump Power}}{\dot{M}_{brine}\, C_{brine}}$$

where

Pump Power = the power input to the secondary fluid pump

Some line heat gain is also expected and was set at 0.25°F for the supply and return lines of the loop.

Once the total temperature rise of the secondary fluid loop is determined, the load on the chiller evaporator can be found by

$$Q_{evap} = \dot{M}_{brine}\, C_{brine} \Delta T_{brine}$$

where

ΔT_{brine} = the total temperature gain seen in the fluid

Mechanical subcooling is used in the secondary loop system for the low temperature chiller system in the same fashion as is seen in multiplex systems. A portion of the medium temperature system provides the subcooling. The refrigeration load associated with the subcooling must be added to the total load of the medium temperature system. The subcooling load is calculated based upon the refrigeration load of the low temperature system, which sets the flow rate of refrigerant needed. The subcooling reduces the temperature of the refrigerant from the temperature leaving the condenser to 40°F. The subcooled liquid temperature is factored into the available capacity of the low temperature system in meeting its refrigeration load.

The state points of the chiller system are then determined. The evaporator temperature of the chiller heat exchanger is set at 7°F below the outlet temperature of the fluid loop (either –20 or 20°F). The outlet temperature of the refrigerant is set at 8°F higher than the evaporator temperature. The temperature rise is due primarily to the control action of the thermostatic expansion valve of the chiller heat exchanger, which regulates the outlet temperature of the refrigerant at a superheated condition. Pressure drop of the refrigerant vapor to the suction of the compressors is negligible due to the close-coupling of the compressors and the heat exchanger. The SDT of the compressor system is determined from the condensing temperature, which is calculated depending upon the type of heat rejection system analyzed. The secondary loop refrigeration system can be modeled with air-cooled, water-cooled, or evaporative condensing. The method of determining the condensing temperature for each heat rejection type is the same as described previously.

The central chiller uses multiple parallel compressors to address the chiller heat exchanger load. The types of compressors now employed in presently installed systems are either screw or reciprocating. Scroll compressors can be analyzed if desired. The procedure for determining the number of compressors operating is the same as that used for the multiplex and distributed systems. Manufacturer's performance data are used to determine compressor cooling capacity and power. The number of compressors operating is found from the ratio of the chiller cooling load to the total available compressor capacity. Compressor energy is the product of the compressor power, number of compressor operating, and the number of hours in the temperature bin being examined.

The energy consumption of the secondary fluid pump is determined using the method outlined previously. While the fluid flow to the display cases varies with changing refrigeration load, the total fluid flow across the pump is constant. Variation in flow is achieved by flow bypassing around the pump. The head addressed by the pump was estimated based upon the temperature of the secondary fluid loop, the secondary fluid employed, the length and diameter of pipe between the cases and chiller, and the pressure drop at the display cases and through the chiller heat exchanger. The secondary fluids examined here consist of propylene glycol/water for the medium temperature loop, and Pekasol 50 for the low temperature loop. Loop piping diameter was sized to maintain a velocity of 4 to 6 ft/sec, while the typical length of piping was estimated at 250 ft. Values of the pressure drop for the display cases range from 5 to 7 ft (WC), while the pressure drop of the chiller heat exchanger was set at 20 ft (WC). The head requirements calculated for the secondary fluid loop pump were 50 and 75 ft (WC) for the medium and low temperature systems, respectively. The pump and motor efficiencies were taken at 55 and 85 percent, respectively.

The energy requirement for heat rejection is dependent upon the type of heat rejection employed. The energy is calculated using the procedures described previously.

5.3 Heat Reclaim for Multiplex Refrigeration

Heat reclaim refers to the use of the reject heat from the refrigeration system for store space heating. Heat reclaim is accomplished by routing the discharge gas from the compressors to a coil located in one or more of the HVAC air handlers. In the coils heat is removed from the refrigerant and is transferred to the store circulation air to provide heat.

The amount of heat that can be recovered in this fashion is determined by the amount of condensing allowed to take place in the heat reclaim coils. Previous attempts to recover a large portion of the heat by complete condensing during heat reclaim resulted in the need for a very large refrigerant charge, since refrigerant was needed to fill both the condenser and the piping between the heat reclaim coils and the condenser. The so-called “winter” refrigerant charge was often larger than could be held in the system during non-heating periods and venting of refrigerant was necessary when the heating season had concluded.

To avoid this excessive amount of refrigerant, the heat reclaim coils are purposely undersized to limit the amount of heat removal. The coils are often sized at a large temperature difference (condensing temperature – inlet air temperature) so that only desuperheating or a very limited amount of condensing occurs.

The amount of heat reclaimable through desuperheating can be estimated from rated compressor data and a cycle calculation as is illustrated in Figure 5-2. The total heat rejection is a combination of the refrigeration capacity and the compressor power. The cycle calculation allows the refrigeration effect and refrigerant mass flow to be found for the rated cooling capacity. The discharge enthalpy, h_{disch}, from the compressor is then determined from

$$h_{disch} = h_{ret} + \frac{\dot{W}_{comp}}{\dot{m}}$$

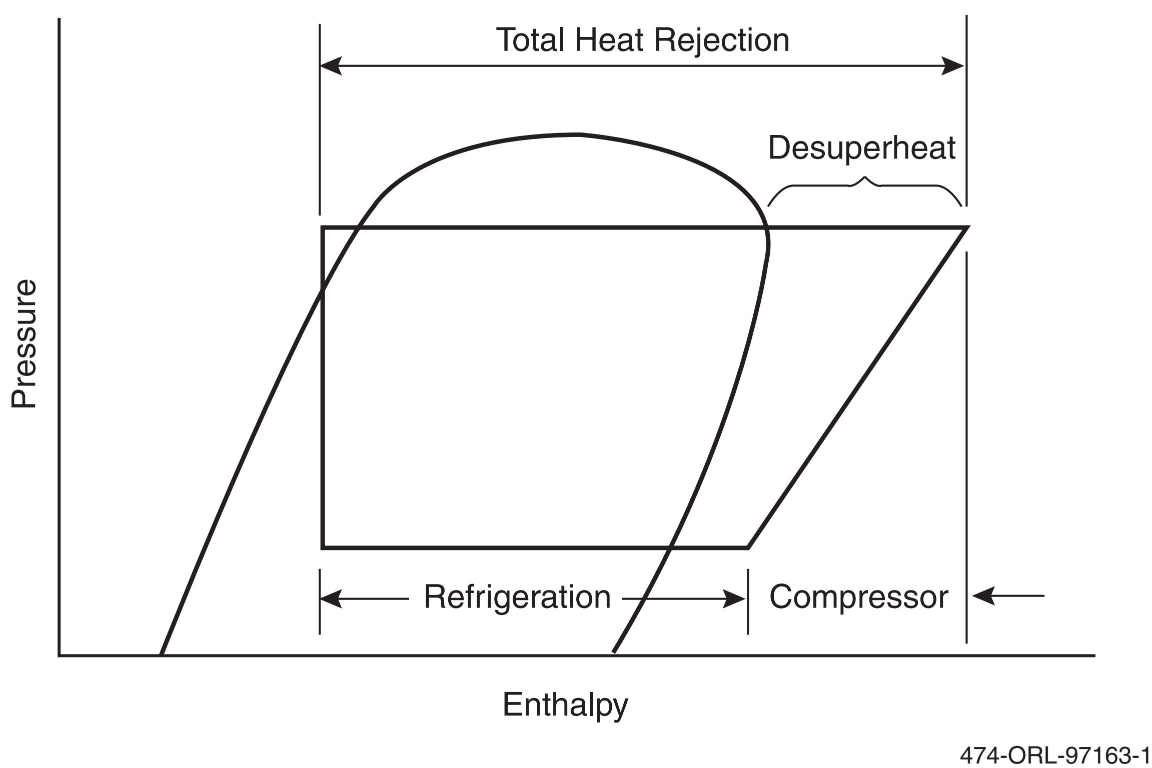

Figure 5-2. Cycle analysis for heat reclaim

where

h_{ret} = the enthalpy of the return gas entering the compressor
$\dot{W}_{comp}$ = the power input to the compressor
$\dot{m}$ = the mass flow rate of refrigerant

The amount of heat recovered, Qrec, by desuperheating is the heat energy contained in the refrigerant greater than that associated with the saturated gas enthalpy.

$$Q_{rec} = \dot{m}\left(h_{disch} - h_g\right)$$

where

h_g = the saturated gas enthalpy of the refrigerant (Btu/lb)

The fraction of heat that is recoverable from desuperheating is

$$\frac{Q_{rec}}{THR}$$

where

THR = total heat rejection
= Refrigeration Capacity + Power

The recovery fractions were evaluated for medium and low temperature refrigeration as a function of evaporator and condenser temperatures, and the return gas temperature was set at 65°F. The refrigeration capacity and compressor work were found from representative compressor performance curves. The following relations were found for the heat recovery fraction as a function of condensing temperature.

For medium temperature refrigeration, the recoverable fraction =

0.23 + (2.0e-05*SST – 0.001)*SST + (2.27e-05*SDT - 0.003)*SDT

For low temperature refrigeration, the recoverable fraction =

0.29 + (0.0001*SST +0.002)*SST + (4.36e-05*SDT – 0.03)*SDT

where

SST = the saturated suction temperature (°F)
SDT = the saturated discharge temperature (°F)

Based on this calculation, it can be found that the amount of heat recovered is small compared to the total heat of rejection. For low temperature refrigeration the fraction of heat recovered ranges from 0.31 to 0.38 for condensing temperatures of 70 and 90°F, respectively. The fraction for medium temperature refrigeration is smaller at 0.16 to 0.24 for these same condensing temperatures.

The standard method for heat reclaim coil sizing consists of allowing a maximum heat recovery of 50 percent of the total heat rejection at a temperature difference (condensing temperature – inlet air temperature) of 35 to 40°F. This sizing is made at the design conditions so that the amount of heat recovered during operation is actually substantially less primarily because the temperature difference seen at the heat reclaim coil ranges from 5 to 30°F, depending upon the value of minimum condensing temperature applied to the refrigeration. The highest value of minimum condensing temperature normally seen is on the order of 95°F. At this condition, partial condensing of the refrigerant will occur to a quality no less than 90 percent. For analysis of heat reclaim performance, the amount of heat recovered can be estimated as being proportional to the temperature difference seen at the heat reclaim coil.

Fan power requirements associated with the heat reclaim coils are also significant in terms of total annual energy consumed. Air must flow through coils at all times, therefore, fan power is consumed regardless whether space heating is needed or not. A typical pressure drop for the heat reclaim coil is on the order of 0.2 in. WC. The flow rate of air passing through the coil must be capable of maintaining a coil face velocity of 500 ft/min. For the application considered here, the required flow is 31,250 cfm. Using a fan efficiency of 0.2, yields a fan power of 3.4 kW. Annual energy consumption for airflow through the heat reclaim coil is 29,784 kWh.

Energy analysis of heat reclaim consists of first determining the annual energy consumption of the refrigeration system at the minimum condensing temperature chosen for heat reclaim operation. At each ambient temperature the amount of heat recovered is calculated based on the expected TD of the heat reclaim coil. For this calculation, the amount of heat recovered is expected to be proportional to the TD. The recovered heat is then compared to the expected building heating load at this same ambient temperature. Only a fraction of the recovered heat will be utilized if the space heat load is less than the amount of heat reclaim. It is also possible that the heating load will exceed the amount of heat recovered, which requires the use of auxiliary heating to satisfy the heating load. A comparison of the combined energy consumption for both space heating and refrigeration can be made to determine the benefit derived from heat reclaim.

5.4 Supermarket HVAC Analysis

5.4.1 HVAC Model Description

The model used for the analysis of supermarket HVAC employs the same ambient temperature bin method as the refrigeration model to determine annual energy use. Figure 5-3 shows a diagram of the model and major steps followed.

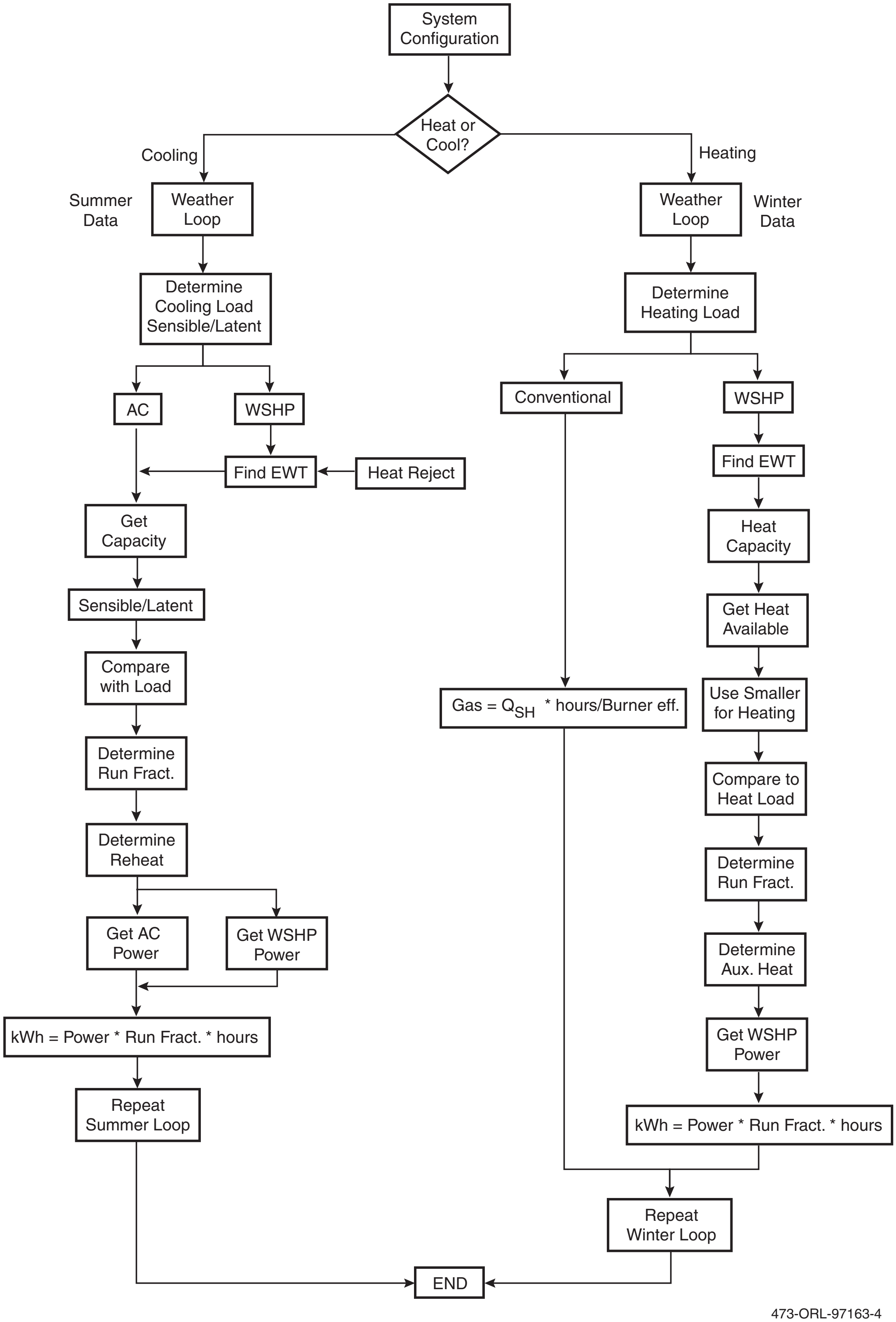

Figure 5-3.   HVAC model flow chart

The HVAC model uses a configuration file to initiate the analysis. The configuration file provides the following information:

- Type of HVAC system to be analyzed – conventional or water-source heat pump.
- Space heating or cooling analysis.
- Temperature set point for the store.
- Humidity set point for the store (specified as %RH).
- Installed capacity for space cooling (tons).
- Ventilation flow rate (specified as percent of total store air flow).

The installed space cooling capacity relates to the evaluation of either standard air conditioning or water-source heat pump. The cooling capacity also determines the heating capacity of the water-source heat pump.

The model then determines if space heating or cooling analysis is desired. The type of weather data employed for the analysis is also set by this parameter. For space heating, weather data occurring in the six months of November through April (winter data) are considered. The space cooling analysis examines weather data for May through October (summer data).

The next step in the space cooling analysis is to determine the space cooling load of the store, including both the sensible and latent components. The procedure used is presented in detail in the description of the modeled supermarket to find the design cooling load. Several exceptions to this procedure are used when determining the cooling load during non-design conditions. The sensible and latent loads due to people are reduced to reflect an occupancy of 70, rather than 200 (design value), people. Solar loads through roof and glass store fronts are also reduced to account for lower solar radiation levels and nighttime operation.

The sensible and latent cooling capacities of the installed air conditioning system for the ambient conditions of the bin are now determined. For the conventional air conditioner, capacity is determined based upon the ambient air temperature and the air dry-bulb temperature entering the cooling coil (T_{edb}). T_{edb} is found by considering the mix temperature of the return and ventilation air flows. For the water-source heat pump, total cooling capacity is related to the water temperature entering the heat pump condenser (T_{we}), and the T_{edb} at the cooling coil. T_{we} is determined by the heat rejection employed, which is normally an evaporative fluid cooler. The typical water temperature approach to the ambient wet-bulb is 15°F. The latent component of the cooling capacity is found based upon the entering wet-bulb temperature (T_{ewb}) to the cooling coil. The sensible component is taken as the difference between the total and latent capacities.

The run fraction for the space cooling system is found by comparing the latent and sensible parts of the cooling load to the corresponding components of the cooling capacity. The load-to-capacity ratios for sensible and latent are compared and the larger of the two is considered the run fraction of the cooling system.

In the instance when the run fraction is set by the latent load, excess sensible cooling of the store will occur and reheat is applied to maintain the set point temperature. The amount of reheat needed is determined from the difference between the amount of sensible cooling supplied and the sensible cooling load. The amount of cooling applied is the product of the sensible cooling capacity and the system run fraction. The analysis assumes that the reheat is supplied by a gas duct heater. The gas consumption is determined based on a burner efficiency of 80 percent.

The power and energy needed for space cooling is found from the energy efficiency ratio (EER) of the cooling system. Both the conventional air conditioner and the water-source heat pump have EERs at standard rating conditions. The rated EER can be modified for non-rated conditions. For the conventional air conditioner, the EER is a function of the ambient air dry-bulb temperature. The EER of the water-source heat pump is determined from a relation with the entering water temperature. The power requirement for the temperature bin is the product of the EER and the total cooling capacity. The energy used is the product of the power, the system run fraction, and the number of hours in the bin.

For space heating analysis, winter weather data are used to determine the space heating load for each temperature bin. The procedure is the same as that used for finding the design space heating load, but using the bin's dry-bulb temperature, and allowing for the average solar load and building occupancy.

For conventional system, space heating is provided by a gas-fired duct heater. The gas consumption is the product of the space heating load, the number of hours in the bin, and the burner efficiency, which is set at 80 percent.

For analysis of space heating with the water-source heat pump, the initial step is to determine the amount of heat that can be removed from the water loop. The heat absorbing capacity can be estimated from the rated cooling capacity and is approximately 97 percent of the rated cooling at an entering water temperature of 85°F. The analysis uses the installed cooling tonnage to determine the rated cooling by applying a correction for the design entering water temperature. The absorbed heat capacity, Q_{abs}, at rating is then corrected for the actual entering water temperature for the temperature bin. This corrected value is checked against the amount of rejected heat produced by the refrigeration system, which is transferred to the water loop. If the absorbed heat capacity is larger than the refrigeration reject heat, the absorbed capacity is reduced to match the reject heat and the entering water temperature is recalculated to the value corresponding this heat absorption. The total heat capacity, Q_{tot}, provided by the heat pump for space heating is

$$Q_{tot} = Q_{abs} * \left(1 + \frac{1}{(COP_H - 1)}\right)$$

where

Q_{abs} = absorbed heat capacity
COP_H = heating coefficient of performance

A rated heat pump COP_H is specified for an entering water temperature of 85°F. A correction factor is applied for the entering water temperature calculated for the temperature bin.

The total heating capacity of the heat pump is compared to the space heating load. If the capacity is greater than or equal to the heating load, a run fraction for the heat pump is found from the ratio of the space heat load to the heating capacity. If the space heating load is greater than the heating capacity, the run fraction of the heat pump is set at 1 and the amount of auxiliary heat that must be supplied is determined. The auxiliary heating is equal to the difference between the space heating load and the total heating capacity.

The power used by the heat pump is found from the relation

$$\text{Heat Pump Power} = \frac{Q_{abs}}{\left(COP_H - 1\right)}$$

The energy consumed by the water-source heat pump is the product of the power, the heat pump run fraction, and the number of hours in the temperature bin.

The auxiliary heat can be supplied by either gas or electric duct heaters. For the gas heaters, a burner efficiency of 0.8 is used to determine the amount of gas consumed.

The calculations are repeated for all temperature bins contained in the weather data file.

5.4.2 Analysis of Conventional Air Conditioning Systems

Modeling and performance estimates of air conditioning systems are required to evaluate the energy consumption of supermarket HVAC systems. Data from air conditioner manufacturers are available to allow such predictions. The information employed here was found in the Carrier Product Catalog (5-5) for standard rooftop packages, which contain all components needed for store HVAC. The rooftop package will have a standard vapor compression air conditioning system with compressor, condenser and evaporator. The rooftop package also has a blower and air handler to pass store air through the air conditioner evaporator. The air handler provides a duct and damper to control the addition of ventilation air from the outside, which is mixed with the return air from the store at the suction side of the blower. The air is then pumped by the blower through the evaporator coil and discharged to the store. The rooftop unit also provides store heating by adding a gas-fired heater to the supply-side ducting.

For the supermarket air conditioning system, the ability to address the sensible and latent components of the cooling load must be considered. The load split of the air conditioner can be characterized through the concept of the bypass factor (BF) of the evaporator coil. The bypass factor represents the fraction of air that passes through the coil that is unaffected by the coil's cooling. The remainder of the air is cooled to the surface temperature of the coil, which is referred to as the apparatus dew point (ADP) temperature. The BF of a coil is controlled by the amount of air passing through the coil with the BF increasing as the air flow rate is increased.

Lower air flow produces lower values of BF, and larger latent cooling capacity. Higher values of BF increase the total cooling capacity of the air conditioner. The value of BF typically ranges between 0.03 to 0.13.

The split between sensible and latent cooling is also influenced by the entering wet-bulb (T_{ewb}) and dry-bulb (T_{edb}) temperatures. Rating data shows values of total and sensible cooling capacities at three values of T_{ewb} of 62, 67, and 72°F, and at a T_{edb} of 80°F.

Ambient air temperature also has some effect on the cooling capacity of the air conditioner and a strong effect on compressor power, due to changes in condensing temperature.

The rated data for a typical air conditioner were used in a multi-variable regression analysis to determine the values of total and sensible cooling capacities, and compressor power as functions of BF, T_{ewb}, and ambient temperature (T_{amb}). The following relations were determined:

$$\text{Total Cooling Capacity, } Cap_{tot}\text{ (kBtuh)} = 82.897 + 971.83*BF + 5.6357*T_{ewb} - 1.6578*T_{amb}$$

$$\text{Sensible Cooling Capacity, } Cap_{sens}\text{ (kBtuh)} = 974.72 + 1376.8*BF - 8.8357*T_{ewb} - 1.993*T_{amb}$$

$$\text{Compressor Power (kW)} = -12.711 + 50.694*BF + 0.2931*T_{ewb} + 0.238*T_{amb}$$

The sensible cooling capacity is also affected by T_{edb}. The catalog gives the following relation to correct the sensible cooling capacity for T_{edb} values other than 80°F.

$$\text{Sensible Cooling Correction Factor} = 1.10*(1 - BF)*(T_{edb} - 80)$$

The sensible cooling correction factor is added to the above sensible cooling capacity to determine the sensible cooling capacity at the actual value of T_{edb}.

Once the total and sensible cooling capacities are found, the latent capacity is determined from

$$\text{Latent Cooling Capacity, } Cap_{lat} = Cap_{tot} - Cap_{sens}$$

The above relations can be used to estimate the energy consumption of the air conditioning through the following procedure. For a given ambient condition (dry-bulb and coincident wet-bulb temperatures), the space cooling load for the supermarket can be estimated, along with the sensible and latent load components. Given the store set points for temperature and humidity, and the ventilation rate, the temperature and humidity of the air entering the air conditioner coil can be found. The above performance equations are used to find the sensible and latent cooling capacities of the air conditioner. The required run fraction of the air conditioner is found by

$$\text{Sensible Run Fraction, } RF_{sens} = \frac{Q_{sens}}{Cap_{sens}}$$

$$\text{Latent Run Fraction, } RF_{lat} = \frac{Q_{lat}}{Cap_{lat}}$$

where

Q and Cap represent the load and capacity, respectively
sens and lat refer to the sensible and latent components

The run fraction used for energy consumption estimate is the larger of the two and the compressor power is found from the relation shown above

Air Conditioner Energy = RF * Air Conditioner Power * hours

where

Air Conditioner Power = Compressor Power + Condenser Fan Power (1.7 kW)
Hours = number of hours at the ambient conditions specified

The above procedure also allows the estimate of reheat requirements. Reheat will occur when the latent run fraction is greater than the sensible. In this situation, more sensible cooling will be provided than is needed to satisfy the sensible load. The amount of reheat needed can be estimated from

Reheat required = $Cap_{sens} * RF_{lat} - Q_{sens}$

5.4.3 Water-Source Heat Pump

The water-source heat pump (WSHP) can supply either heating or cooling to the supermarket sales area and works in conjunction with the glycol/water loop used for refrigeration system heat rejection. The fluid loop provides heat rejection when the WSHP is space cooling, or can be the source of heat for the heat pump when store space heating is called for.

Analysis of WSHP requires that the heating and cooling performance of the heat pump be known in relation to ambient conditions inside and outside the store. The heating and cooling capacity of the heat pump are given in manufacturer's data in terms of the entering water temperature (T_{we}). Power requirement for the heat pump is found from the value of the EER, for cooling, and COP_H, for heating. Both of these quantities are given in the literature as functions of T_{we}.

Cooling performance of the WSHP is also characterized by the inlet air temperature and humidity entering the cooling coil, and the bypass factor of the coil. These relations are similar to those presented previously for the conventional air conditioner and are used here to determine the split between latent and sensible cooling for space cooling analysis.

Manufacturer's data (5-5) were used to establish the relation between the entering water temperature and the heating and cooling capacity of the WSHP. These data were correlated using standard linear regression procedures. In order to use these relations for any size heat pump examined, the resulting equations were normalized in terms of the rated heating and cooling capacities of the heat pump. The rated heating capacity of the heat pump occurs at a T_{we} of 70°F, while the rated cooling capacity is at a T_{we} of 85°F.

The relation for space cooling capacity of the WSHP with change in T_{we} is the following

$$\frac{Cap}{Cap_r} = -0.0061T_{we} + 1.5214$$

where

Cap = the total cooling capacity
Cap_r = the total cooling capacity at the rated condition

The EER of the heat pump in cooling mode can be expressed as follows

$$\frac{EER}{EER_r} = -0.0127T_{we} + 2.0936$$

where

EER = the energy efficiency ratio of the heat pump
EER_r = the energy efficiency ratio at the rated condition

The space heating capacity of the heat pump is described in terms of the total heat absorbed (Q_{abs}) from the water loop and the COP. The total amount of heat delivered to the conditioned space, Q_{tot} is the sum of Q_{abs} and the power consumed by the compressor, which is found from

$$Q_{tot} = Q_{abs}\left(1 + \frac{1}{(COP_H - 1)}\right)$$

The relation found for the ratio of $\frac{Q_{abs}}{Q_{absr}}$ and T_{we} is

$$\frac{Q_{abs}}{Q_{absr}} = 0.004T_{we} + 0.6789$$

The heat pump COP_H can be found from

$$\frac{COP_H}{COP_{Hr}} = 0.0022T_{we} + 0.8467$$

Section 5 References

5-1. *Engineering Weather Data,* Departments of the Air force, the Army, and the Navy, Washington, DC, July 1, 1978.

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5-3. Walker, D.H., Foster-Miller, Inc., “Development of an Evaporative Condenser for Northern Climates,” Report prepared for Niagara Blower Co., Buffalo, NY., April, 1997.

5-4. Borton, D.N., and D.H. Walker, “Development of a Demand Defrost Controller,” New York State Energy Research & Development Authority, Contract No. 800-CON-BCS-86, March, 1992.

5-5. *Carrier Products and Systems 1992/1993 Master Catalog,* United Technologies Carrier.

6. DESCRIPTION OF THE MODELED SUPERMARKET

A description of a typical, modern supermarket was constructed for the analysis of various refrigeration and HVAC systems. The conditioned sales area for this modeled market was set at 40,000 ft^2, which is a representative size for the majority of stores now in operation or recently constructed.

6.1 Refrigeration Description

A typical refrigeration load schedule was devised for modeling and comparison of the refrigeration systems examined. The refrigeration schedule describes the connected refrigerated fixtures, made up of display cases and walk-in storage coolers, and gives the desired discharge air and evaporator temperatures, and the refrigeration load at design conditions (75°F, 55% RH). Table 6-1 provides a listing of all refrigerated fixtures in the modeled supermarket.

The refrigeration loads were then assigned to the multiplex, distributed, and secondary loop systems so that a refrigeration system description was formulated for each type of refrigeration system examined. The resulting configurations are shown in Tables 6-2, 6-3, and 6-4 for the multiplex, distributed, and secondary loop systems, respectively. For the multiplex system, the refrigeration loads are assigned to the compressor racks based upon the evaporator temperature of each load and saturated suction temperature (SST) of the racks. The load is assigned to the rack with the SST closest to, and less than the evaporator temperature of the load. Table 6-2 shows that several of the compressor racks employ two SST levels, which signifies that a split suction manifold is used with a common discharge for both suction levels. System performance analysis is done at each SST value.

Table 6-3 shows the refrigeration load configuration for the distributed refrigeration system. The compressor cabinets are matched to the refrigeration loads in the same fashion as is used for the multiplex system. Because the compressor cabinets are located throughout the store, less refrigeration circuits are assigned to each cabinet, and usually one compressor cabinet is used for refrigeration of a particular product, such as meat, produce, or frozen food. For this reason, the match between the SST of the compressor cabinets and the evaporator temperature can be kept closer than is seen in multiplex systems. If more than one SST is needed in a particular locality in the store, a split suction manifold can also be used in a compressor cabinet. Analysis of the distributed system is also done for each SST value seen in the supermarket.

Table 6-4 shows the arrangement of refrigeration loads for the secondary loop system. The secondary loop system operates at four secondary fluid loop temperatures of –20, 10, 20 and

Table 6-1. Description of modeled cases and coolers (low temperature)

System No.	Case or Cooler Description	Length No. of Doors or Floor Size (ft)	Discharge Air Temperature (°F)	Evap Temperature (°F)	Design Load (Btuh)
1	Ice Cream Door Cases	17 doors	-12	-19	26,180
2	Ice Cream Door Cases	17 doors	-12	-19	26,180
3	Ice Cream Door Cases	19 doors	-12	-19	29,260
4	Ice Cream Door Cases	20 doors	-12	-19	30,800
5	Frozen Meat	28	-10	-20	12,100
6	Grocery Freezer	42 x 15	-10	-20	39,900
7	Frozen Fish	12	-12	-19	3,300
9	Frozen Food Door Cases	15 doors	-5	-11	21,375
10	Frozen Food Door Cases	16 doors	-5	-11	29,300
11	Frozen Food Door Cases	15 doors	-5	-11	21,375
12	Frozen Food Door Cases	16 doors	-5	-11	22,800
13	Bakery Freezer	12 x 10	-5	-11	9,800
14	Deli Freezer	8 x 10	-5	-11	7,000
16	Meat Cases	40	24	15	54,800
17A	Meat Cases	36	22	17	15,120
17B	Fish Cases	12	22	17	12,660
18	Meat Cases	30	24	18	12,580
19A	Bakery Cooler	6 x 8	28	18	10,560
19B	Deli Cases	20	32	18	29,900
19C	Deli Cases	8	30	25	3,360
19D	Deli Cases	16	26	20	7,040
19E	Deli Cooler	12 x 11	36	29	9,100
19F	Cooler	6 x 8	25	15	10,560
19G	Cheese Cooler	6 x 14	25	15	18,480
20A	Deli Cooler	6 x 8	25	15	10,560
20B	Deli/Meat Cases	32	32	18	46,880
21	Meat Cooler	40 x 15	30	22	35,800
22	Meat Box	15 x 10	30	22	11,400
24A	Dairy/Deli Cases	36	32	21	54,000
24B	Dairy/Deli Cases	40	32	21	60,000
25A	Dairy	12	32	18	18,000
25B	Beverage	36	32	18	53,820
25C	Beverage	24	32	18	36,000
26A	Produce	24	32	21	35,880
26B	Produce	56	39	21	43,120
26C	Produce	82	39	21	62,440
26D	Floral Cooler	12 x 11	38	32	11,675
26E	Floral Case	13		21	12,350
27A	Bakery Retarder	12 x 8	36	28	7,300
27B	Dairy Cooler	40 x 20	36	28	54,125
28A	Meat Prep	40 x 15	50	35	74,250
28B	Produce Cooler	36 x 14	38	30	27,200

Table 6-2. Multiplex refrigeration system configuration for the modeled supermarket

	SST (°F)	Design Refrigerant Load (Btuh)	Refrigerant
Low Temperature			
Rack A	-23	167,720	R404A
Rack B	-14	111,650	R404A
Total Low Temperature Load		279,370	
Medium Temperature			
Rack C	13	222,100	R-22
	18	66,700	
Rack D	19	375,610	R-22
	26	174,550	
Medium Temperature Load		838,960	
Design Subcooling		53,000	
Subcooling Load varies with change in low temperature refrigeration load			

Table 6-3. Distributed refrigeration system configuration for the modeled supermarket

	SST (°F)	Design Refrigerant Load (Btuh)	Refrigerant
Low Temperature			
Cabinet A	-14	15,400	R-404A
Cabinet B	-23	152,320	
Cabinet C	-14	111,650	
Total Low Temperature Load		279,370	
Medium Temperature			
Cabinet D	19	105,720	R-404A
Cabinet E	14	35,490	
Cabinet F	13	39,600	
Cabinet G	19	262,120	
Cabinet H	19	212,380	
Cabinet I	26	70,525	
Cabinet J	29	113,125	
Total Medium Temperature Load		838,960	

30°F. Four chiller systems are employed, one for each fluid loop. These four loop temperatures are capable of addressing all display case loads, including ice cream and meats (6-4).

The multiplex and secondary loop systems employ mechanical subcooling for the low temperature refrigeration that is supplied by the medium temperature refrigeration. A design subcooling load is shown in the table for both these refrigeration systems. During operation, the amount of subcooling required by the low temperature systems varies with the size of the low temperature load. As part of the analysis, the subcooling load is calculated for each temperature bin and is added to the medium temperature refrigeration load. The subcooling load is not included in the total medium temperature refrigeration shown for the multiplex and secondary loop systems so that the loads can be compared with the medium temperature load of the distributed refrigeration. Subcooling for the low temperature refrigeration in the distributed

Table 6-4. Secondary loop refrigeration system configuration for the modeled supermarket

	Loop Temperature (°F)	Design Refrigerant Load (Btuh)	Refrigerant
Low Temperature			
Total for Low Temperature Loop	-20	279,370	R-507
Medium Temperature			
MT-1	10	75,090	R-507
MT-2	20	580,220	
MT-3	30	183,650	
Total for Medium Temperature Loop		838,960	R-507
Design Subcooling Load		53,000	
Subcooling load varies with change in low temperature refrigeration load			

systems is provided by vapor injection at the low temperature compressors. The capacity and power changes seen at the compressors are included as part of the manufacturer performance data, and is therefore, not specified as a particular load.

Table 6-5 lists the heat rejection methods employed for each system. The choice of heat rejection was made on the basis of most common practice for each system type. The multiplex and secondary loop systems are modeled with air-cooled condensers that are sized at TDs (condenser – dry bulb) of 10 and 15 °F, for low and medium temperature refrigeration, respectively. The distributed refrigeration uses water-cooled condensers located in each of the compressor cabinets. The TD (condenser – inlet water) for the water-cooled condenser is 10°F.

Table 6-5. Heat rejection for modeled refrigeration systems

			Refrigerant System Applied		
Heat Rejection	Temperature Difference (TD)	TD Value (°F)	Mult.	Dist.	Sec. Loop
Air-Cooled Condenser	Condenser – Ambient Dry-bulb	Low temperature - 10 Medium temperature – 15	x		x
Evaporative Condenser	Condenser – Ambient Wet-bulb	12	x		x
Water-Cooled Condenser	Condenser – Entering water	10	x	x	x
Evaporative Fluid Cooler	Leaving water – Ambient Wet-bulb	12	x	x	x
Dry Fluid Cooler	Leaving water – Ambient Dry-bulb	12	x	x	x

The fluid cooler used to reject heat from the water loop is evaporative and operates at an approach temperature (leaving water – wet-bulb) of 12°F.

The use of the evaporative fluid cooler with the distributed refrigeration is currently the most prevalent method for heat rejection with these systems, since rejection to the wet-bulb temperature helps to overcome the added temperature difference associated with the fluid loop. Dry fluid coolers can also be employed, in which case, a representative approach temperature (leaving water – dry-bulb) would be 12°F. The resulting difference between the refrigeration condensing and ambient dry-bulb temperatures will be higher than is seen with air-cooled condensing, on the order of 22°F, which will hurt the performance of the distributed system. Air-cooled condensers can also be used with the distributed system and were modeled here. Condensing temperatures for a distributed system with air-cooled condensers would be similar to these seen with multiplex with air-cooled condensers.

Operation of the multiplex and secondary loop systems with either evaporative or water-cooled condensers is also possible, but is not considered standard practice. For evaporative condensers, best operation occurs at a TD (condenser – wet-bulb) of 12°F. A water-cooled rejection system could employ either an evaporative or a dry fluid cooler. The condenser and ambient temperature differences would be the same as seen for the distributed refrigeration system.

Table 6-6 lists the remaining system parameters specified for the refrigeration system analysis. The significant parameters are:

- Pressure drop between the evaporators and the compressor suction – expressed as a difference between the saturated evaporator and suction temperatures. The largest drop is seen in the multiplex refrigeration because of the long suction pipe runs between the display cases and the compressor racks. The close coupling of the compressor cabinets and the display cases seen with the distributed system helps to reduce this loss to 2°F or less. For the secondary loop refrigeration, the chiller evaporator is mounted on the same skid as the compressors, which reduces this pressure drop to a very small value; a value of 0 was used in this analysis.

Table 6-6. Parameters used for refrigeration system analysis

				Liquid Refrigerant Temperature (ºF)	
System	System Pressure Drop (SET-SST) (ºF)	Return Gas Temperature (ºF)	Minimum Condensing Temperature (ºF)	No Subcool	Subcooled
Multiplex	3	45	70	Tcon - 10	45
Distributed	2	Tevap + 15	60	Tcon - 10	Variable
Secondary Loop	0	Tevap +10	70	Tcon - 10	40

- Return gas temperature – the rise in suction gas temperature due to ambient heating reduces the mass flow capability of the compressors, which results in longer run times to satisfy the refrigeration load. The return gas temperature was set at 45°F for both low and medium temperature for the multiplex system. For the distributed and secondary loop systems, the value of the return gas temperature was set at 15 and 10°F higher than the saturated suction temperature, respectively.

- Minimum condensing temperature – the lowest condenser temperature value the system is allowed to operate. A value of 70°F was used for the multiplex and secondary loop refrigeration systems, because both employed reciprocating compressors in the present analysis and this is the lowest condensing temperature recommended for these compressors. For the distributed system, the minimum condensing temperature was set at 60°F, because of the use of scroll compressors, which can be operated at lower condensing temperatures than reciprocating units. For the low-charge multiplex system, the minimum condensing temperatures were set at 40 and 60°F for low and medium temperature refrigeration, respectively. The condenser control scheme of the low-charge multiplex allows the compressors to operate at these lower temperature values (6-5).

- Liquid refrigerant temperature – for non-subcooled systems, the liquid temperature was set at 10°F less than the condensing temperature. This value is typical in systems where fan cycling is used for head pressure control. For subcooled systems, the liquid temperature varied according to system type. A value of 45°F was used for multiplex refrigeration to account for temperature rise seen between the subcooler heat exchanger and the display case inlet. The temperature of the liquid leaving the subcooler was estimated at 40°F, which is a typical set point value for mechanical subcooling. The subcooled liquid temperature for the distributed system is variable depending upon the vapor injection flow rate seen at the scroll compressors. In general, the subcooling provided by the vapor injection will reduce the liquid temperature leaving the condenser by approximately 15 to 20°F. Minimal temperature rise is anticipated because of the close proximity of the compressor cabinets to the display cases. No temperature rise is expected for the secondary loop refrigeration because the subcooled refrigerant liquid is used in the low temperature chiller.

Table 6-7 lists system parameters unique to the operation of the secondary fluid loops for secondary loop refrigeration. The fluid temperature difference across the display cases was set at

Table 6-7. System parameters used for the analysis of secondary loop refrigeration

Parameter	Value
Fluid Temperature Difference (ºF)	7
Chiller Approach (Fluid leaving – Evaporator) (ºF)	5
Secondary Fluids	
Medium Temperature	Propylene Glycol/Water
Low Temperature	Pekasol 50

a nominal 7°F. The approach temperature difference for the chillers was set at 5°F. The secondary fluids selected for modeling were a mixture of propylene glycol and water for medium temperature and a potassium formate/water mixture that is marketed under the product name Pekasol 50. The secondary loop refrigeration system was modeled as employing reciprocating rather than screw compressors. Energy related performance of the reciprocating compressors is better than that seen with the screw compressors and the results presented here represent the lowest energy consumption expected from the secondary loop refrigeration.

The effect of defrost on refrigeration system performance was also considered in the analysis. For the multiplex and distributed refrigeration systems, hot gas defrost is employed for low temperature refrigeration. Defrost of the medium temperature system is done through off-cycle defrost. The secondary loop system uses warm glycol for both low and medium temperature refrigeration defrosts.

To evaluate the effect of ambient temperature on refrigeration system performance, four locations were chosen and are listed in Table 6-8 according to the average number of degree-days of heating seen at each location. These sites represent a wide range of ambient conditions with the highest number of degree-days occurring in Worcester, MA and the lowest in Los Angeles, CA.

6.2 HVAC Description for the Modeled Supermarket

Space cooling and heating loads were determined for each of the sites selected using standard methods to calculate these loads. A description of the methods used is provided below.

6.2.1 Space Cooling Load

The air conditioning load for a supermarket is unique because of the large amount of refrigerated fixtures installed, which greatly influence both the sensible and latent components of the cooling load. Internal sensible loads such as lights and other equipment normally dominate air conditioning loads for commercial buildings. Air conditioning in most commercial buildings tends to operate over an extended portion of the year, despite ambient temperatures, which are less than the building's temperature set point. For supermarkets, the sensible heating of the internal loads are negated by the large amount of heat removed by the refrigerated fixtures. The sensible portion of the air conditioning load is substantially smaller, and the total air conditioning load of the supermarket has a large latent load component. Latent loads are also important to the operation of the refrigerated display cases, because ambient moisture is captured by the cases and deposited as frost on the evaporator. Frost loading can reduce air flow through the evaporator, reducing the refrigerating capability of the case. The maximum ambient conditions at which the display case is rated are a dry-bulb temperature of 75°F and a relative

Table 6-8. Locations chosen for supermarket refrigeration analysis

Location	Average Heating Degree-Days
Worcester, MA	6,848
Washington, DC	4,550
Memphis, TN	3,227
Los Angeles, CA	1,819

humidity of 55 percent. These values are often used as the control points for the store air conditioning. Because of the large latent load seen in a supermarket, both a thermostat and humidistat are employed to control the air conditioning. In some locations, it is possible to have a latent load large enough to cause the store air conditioning to operate after the sensible cooling demand has been met. In these situations, reheat of the air may be required to maintain a condition in the store that is acceptable to customers and store employees. Reheat can be done using heat reclaim from the refrigeration if the store's HVAC system is equipped with reclaim coils at the air handlers.

The sizing of the air conditioning system is based upon the design air conditioning load, which can be calculated using the procedures outline in ref. (6-1). Additional specific information concerning HVAC loads for supermarkets can be found in ref. (6-2).

The air conditioning load for a supermarket consists of sensible and latent portions. The sensible cooling load is derived from the following elements:

- Conduction through the building walls and roof – The conduction load can be determined through methods outlined in ASHRAE. For analysis, wall construction was characterized as 8 in. concrete block with insulation. The heat transfer coefficient, U, for this wall type is listed as 0.103 Btu/hr-ft^2-°F. The roof construction was considered to be steel sheet with 2 in. of insulation and a U of 0.092 Btu/hr-ft^2-°F. One wall of the supermarket was taken to be essentially all glass with a U value of 1.0 Btu/hr-ft^2-°F.

- Roof solar loading – Solar insolation, particularly on the roof, can add significantly to the cooling load of the building. The procedure used to determine the roof solar load involved the Cooling Load Temperature Difference (CLTD), described in ASHRAE, which determines an equivalent temperature difference to take into account the solar loading. For each location examined, peak and average CLTD values were calculated. The peak value was used to determine the design cooling load, while the average value was used in energy analysis calculations.

- Glass solar transmittance – A sensible load was included to account for solar gain through the glass front of the supermarket. Peak and average values of 36,000 and 18,000 Btuh were used for design and energy calculations, respectively.

- Fan heating – The power associated with the HVAC fans add to the sensible cooling load. The value of the fan load was taken as a constant 61,000 Btuh.

- Ventilation – The ventilation air flow into the supermarket will have both latent and sensible loads. The sensible load is found from the change in air temperature from outside ambient to store. The flow rate of ventilation air was set at 10 percent of the total store circulation.

- Infiltration – The amount of ambient air entering the store through infiltration was set at 5 percent of total store circulation.

- People – People provide a total cooling load of 450 Btuh, of which 250 Btuh is sensible. The peak number of people in the store was taken to be 200, while the average number of people was set at 70.

- Lighting and miscellaneous heat loads – The lighting level in the store was taken as 3.0 Watts/ft^2, which is a typical value for supermarkets. The remainder of the load caused by installed equipment, such as electrical appliances, ovens, etc., was set at 0.8 Watts/ft^2.

The latent portion of the cooling load can be described in terms of the following components:

- Ventilation – Moisture must be removed from the ventilation air to lower the humidity from ambient to store level.

- Infiltration – Infiltration air will also increase store humidity and is addressed in the same fashion as the ventilation air.

- People – The latent portion of the cooling load is 200 Btuh/person.

- Miscellaneous – A constant moisture load of 70 lb/hr was used to account for all remaining latent load addition to the supermarket.

The remaining elements of the cooling load are the cooling credits assigned to the refrigerated display cases. The cooling credits represent the sensible and latent loads removed from the sales area by the operation of the refrigeration. The cooling credits were calculated based upon the refrigeration schedule used for the analysis of the refrigeration compressor systems. The total display case refrigeration load is 768,270 Btuh. The displays fans, lights, and heaters require a cooling load of 141,533 Btuh for the electric input associated with these items, which leaves a net refrigeration load of 626,737 Btuh. The latent portion of the load accounts for approximately 18 percent of the total case load, which amounts to 138,289 Btub. The remaining sensible load is 488,448 Btuh. A correction was applied to the latent load credit since the air conditioning does not have to freeze the moisture associated with this load. The resulting latent credit is 121,966 Btuh.

The design ambient condition for each site examined consisted of the 1 percent values for the dry-bulb and wet-bulb temperatures. Table 6-9 gives these temperature values for each of the sites considered.

Tables 6-10 and 6-11 give the sensible load elements calculated for each site. Roof and glass loads include both solar and conduction loads. The glass load also contains a solar transmittance component.

Table 6-12 describes the latent load elements estimated for the design air conditioning load.

Table 6-9. Design ambient conditions for the supermarket air conditioning load calculation

Location	Design Point Dry-bulb (°F)	Wet-bulb (°F)	Specific Humidity (lb/lb)
Worcester, MA	88	77	0.0176
Washington, DC	93	78	0.0174
Memphis, TN	98	80	0.018
Los Angeles, CA	93	72	0.012

Table 6-10. Sensible cooling load analysis for supermarket air conditioning

Location	Conduction and Solar			
	Roof	Walls	Glass	Fans
Worcester, MA	267,737	20,414	102,066	61,000
Washington, DC	271,169	28,266	127,476	61,000
Memphis, TN	271,169	36,118	152,886	61,000
Los Angeles, CA	271,169	28,266	127,476	61,000

Table 6-11. Sensible load analysis for supermarket air conditioning

Location	Ventilation	Infiltration	Case Credit	Lighting and Miscellaneous	People
Worcester, MA	56,394	28,197	(488,449)	518,776	50,000
Washington, DC	78,084	39,042	(488,449)	518,776	50,000
Memphis, TN	99,774	49,887	(488,449)	518,776	50,000
Los Angeles, CA	78,084	39,042	(488,449)	518,776	50,000

Table 6-12. Analysis of latent loads for supermarket air conditioning

Location	People	Miscellaneous	Ventilation	Infiltration	Case Credit
Worcester, MA	40,000	75,320	147,197	73,598	(121,966)
Washington, DC	40,000	75,320	143,323	71,662	(121,966)
Memphis, TN	40,000	75,320	154,944	77,472	(121,966)
Los Angeles, CA	40,000	75,320	38,736	19,368	(121,966)

Table 6-13 gives the total estimated cooling loads for design of the air conditioning system. For the final sizing, an extra 20 percent cooling capacity was added to ensure that adequate cooling is available at each of the sites. The use of a safety factor like this is standard procedure.

6.2.2 Space Heating Load

The space heating load for a supermarket is made up of two types of load elements, which are fixed, essentially constant, and ambient-dependent, change as ambient dry-bulb temperature changes.

Table 6-13. Air conditioning unit sizing for supermarkets

Location	Total Cooling Loads (Btuh) Sensible	Latent	Total	Sense/Total	Tons	Added 20% Capacity
Worcester, MA	616,135	214,149	830,284	0.74	69.2	83.0
Washington, DC	685,364	208,339	893,703	0.77	74.5	89.4
Memphis, TN	751,161	225,770	976,931	0.77	81.4	97.7
Los Angeles, CA	685,364	51,458	736,822	0.93	61.4	73.7

Space Heating Load = Fixed Load Elements + Ambient-dependent Load Elements

The fixed load elements consist of the following:

- Lighting and miscellaneous heat loads.
- Fan heating.
- People.

The same values for these quantities applied to space cooling were used for space heating analysis with the exception of the people loading. Only the sensible load portion of 250 Btuh/ person is considered for space heating. All of these load elements generate heat in the store and reduce the amount of heating to the supplied by the HVAC.

The ambient dependent portion of the space heating load consists of:

- Ventilation –The heating load is found from the change in air temperature from outside ambient to store. The flow rate of ventilation air was set at 10 percent of the total store circulation.

- Infiltration – The amount of ambient air entering the store through infiltration was set at 5 percent of total store circulation.

- Wall and roof conduction – The same approach and heat transfer coefficients are used to determine these portions of the space heat load.

The impact of the refrigerated display cases on space heating is to increase the amount of heat that must be provided by the HVAC. The value of this increase was the same as the sensible cooling credit of 488,448 Btuh used for determining the space cooling load.

Table 6-14 lists the 99 percent design dry-bulb temperature values for each of the sites examined (6-3). A design space heating load was found for each site based on these design temperatures. The values of the design loads are also listed in Table 6-14.

Table 6-14. Space heating design ambient temperatures and loads for modeled sites

Location	Winter Design Dry-Bulb Temperature (ºF)	Design Space Heating Load (Btuh)
Worcester, MA	0	1,181,055
Washington, DC	10	1,000,435
Memphis, TN	13	946,249
Los Angeles, CA	37	512,761

6.3 Utility Rates

Table 6-15 provides the utility rates for electric, gas, and water used in the analysis. These rates were obtained from the respective local utility of each location considered.

Section 6 References

6.1. *1989 ASHRAE Handbook, Fundamentals,* Chapter 26, "Air-Conditioning Cooling Load," American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc., Atlanta, GA.
6.2. *1987 ASHRAE Handbook, HVAC Systems and Applications,* Chapter 18, "Retail Facilities," American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc., Atlanta, GA.
6.3. *1989 ASHRAE Handbook, Fundamentals,* Chapter 24, "Weather Data," American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc., Atlanta, GA.
6.4. Personal communications with Mr. Yakov Arshansky, Hill-Phoenix Refrigeration Corporation.
6.5. Enviroguard & Enviroguard II, System Technical Brochure, Tyler Refrigeration Corporation, Niles, MI 49120.

Table 6-15. Utility rates used for the supermarket refrigeration and HVAC analysis

	Local Utility Rates		
Location	Electric ($/kWh)	Gas ($/MMBTU)	Water ($/Mgal)
Worcester, MA	0.092	7.34	2.59
Washington, DC	0.071	7.38	1.37
Memphis, TN	0.064	6.11	2.04
Los Angeles, CA	0.092	6.43	2.27

7. ANALYSIS RESULTS

7.1 Refrigeration Analysis Results

Figure 7-1 and Table 7-1 show the estimated annual electric energy consumption for the refrigeration systems analyzed at each of the locations chosen. The multiplex system with air-cooled condensing is considered the baseline, since it is the most commonly installed configuration now used in supermarkets. The results for the low refrigerant charge systems are shown for different methods of heat rejection. The heat rejection method chosen for each system was based upon lowest energy consumption obtained for that particular system. The analysis results show that the lowest energy consumptions were achieved by the distributed systems for operation in Worcester, MA and Washington, DC. Lowest energy consumption was seen for the secondary loop system with evaporative condensing in Memphis, TN and Los Angeles, CA. The low-charge multiplex air-cooled had lower energy consumption than the baseline multiplex system, but also had an energy consumption greater than distributed or secondary loop for all locations. The use of evaporative condensing with the low-charge multiplex lowered the energy consumption, particularly for Memphis, TN and Los Angeles, CA, where this configuration

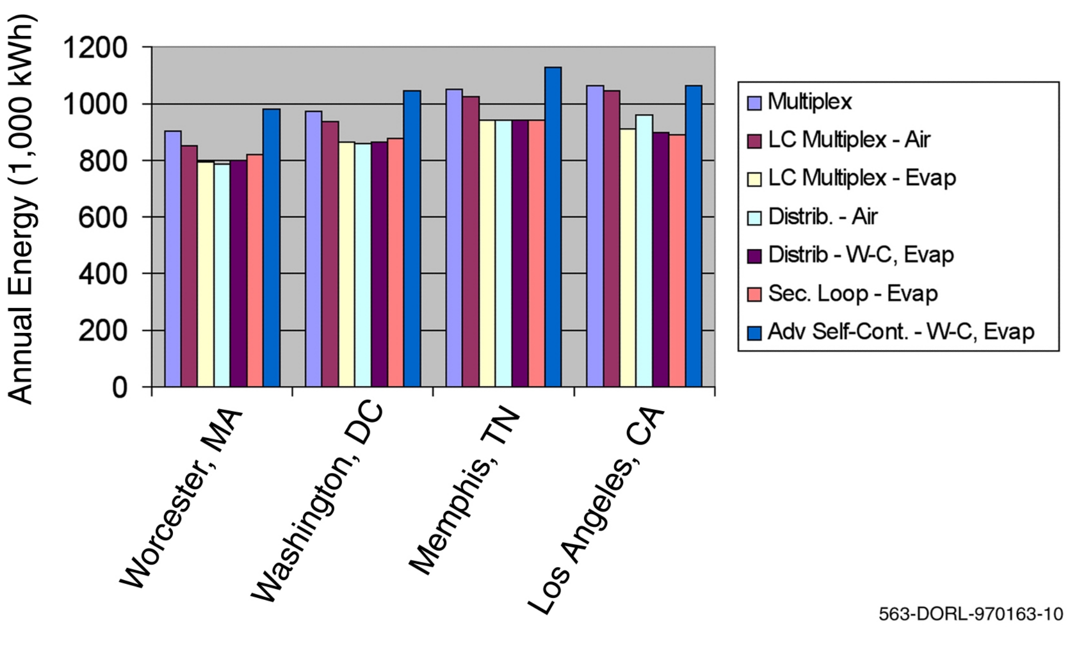

Figure 7-1. Annual energy consumption for low-charge supermarket refrigeration systems

Table 7-1. Annual energy consumption (kWh) for low-charge refrigeration systems for selected locations

Location	Multiplex Air-Cooled	Low-Charge Multiplex Air-Cooled	Low-Charge Mutiplex Evaporative Condenser	Distributed Air-Cooled	Distributed Water-Cooled, Evaporative	Secondary Loop Evaporative Condenser	Advanced Self-Contained Water-Cooled, Evaporative
Worcester, MA	904,500	850,000	791,600	785,700	802,200	821,600	983,700
Washington, DC	976,800	935,200	863,600	860,500	866,100	875,200	1,048,300
Memphis, TN	1,050,200	1,027,100	941,500	942,800	943,200	940,400	1,126,800
Los Angeles, CA	1,067,200	1,042,600	911,300	958,432	894,400	892,400	1,066,800

produced the lowest energy consumption. The advanced self-contained system produced the highest energy consumption for all locations.

Water consumption for evaporative heat rejection was also estimated. The annual consumption of water at each site is given in Table 7-2. Similar amounts of water consumption were seen for each of the refrigeration systems when either evaporative condensing or closed-loop water cooling with evaporative rejection was employed.

Table 7-2. Annual water consumption for heat rejection distributed refrigeration with an evaporative fluid cooler

Location	Annual Water Consumption (gal)
Worcester, MA	1,128,067
Washington, DC	1,179,911
Memphis, TN	1,241,584
Los Angeles, CA	1,225,884

Table 7-3 gives the annual energy savings achieved by each of the low-charge refrigeration systems when compared to the multiplex refrigeration system for operation in Washington, DC. The largest energy savings were achieved by the distributed system operating with air-cooled condensing at 11.9 percent. Similar savings were also seen with the distributed system employing water-cooled condensing and evaporative heat rejection, and with the low-charge

Table 7-3. Energy savings achieved by low-charge refrigeration systems

System	Heat Rejection	Annual Energy (kWh)	Energy Savings versus Multiplex (kWh)	% Savings versus Multiplex
Multiplex	Air-Cooled Condenser	976,800	-	-
Low-Charge Multiplex	Air-Cooled Condenser	935,200	41,600	4.3
Low-Charge Multiplex	Evaporative Condenser	863,600	113,100	11.6
Distributed	Air-Cooled Condenser	860,500	116,300	11.9
Distributed	Water-Cooled Condenser, Evaporative Rejection	866,100	110,700	11.3
Secondary Loop	Evaporative Condenser	875,200	101,600	10.4
Advanced Self-Contained	Water-Cooled Condenser, Evaporative Rejection	1,048,300	-	-
Secondary Loop	Water-Cooled Condenser, Evaporative Rejection	959,700	17,100	1.8

Results for supermarket at Washington, DC location multiplex system using evaporative condensing. Energy savings achieved by these two systems were 11.3 and 11.6 percent, respectively. The secondary loop system with evaporative condensing showed savings of 10.4 percent. Secondary loop refrigeration with water-cooled condensing and evaporative heat rejection, and the advanced self-contained system showed energy consumptions greater than that of the multiplex baseline system.

Table 7-4 gives a breakdown of the annual energy consumption of the refrigeration systems for operation in Washington, DC. Compressor energy consumption is lower for low-charge multiplex, distributed and secondary loop systems. The savings can be attributed to the close-coupling employed by the distributed and secondary loop systems and the lower condensing temperature at which the low-charge multiplex can operate. The distributed and secondary loop systems can also operate at lower condensing temperature when evaporative heat rejection is employed. Energy consumption for the secondary loop system is higher than that of the distributed system, because of the added energy needed for secondary loop pumping. Compressor energy consumption for the advanced self-contained system is significantly higher than that of the multiplex system, despite the close proximity of the compressors to the case evaporators and the use of a minimum condensing temperature of 60°F. The possible explanation for this increase is the use of unloading capacity control for the scroll compressors.

7.1.1 Impact of Heat Rejection on Energy Consumption

The initial results presented showed the refrigeration system energy consumption for each system in its most energy-efficient configuration. Additional analysis was performed to determine the impact of different heat rejection approaches on the energy consumption of the each refrigeration system type. Annual energy consumption estimates are presented in Tables 7-5, 7-6, and 7-7 for multiplex, distributed, and secondary loop refrigeration, respectively. For the multiplex and secondary loop systems air-cooled, evaporative, and water-cooled

Table 7-4. Breakdown of refrigeration annual energy consumption (kWh)

System	Compressors	Secondary Loop Pumps	Heat Rejection*	Total
Multiplex, Air-Cooled	809,400	-	167,400	976,800
Low Charge Multiplex, Air-Cooled	767,700	-	167,400	935,100
Low-Charge Multiplex, Evap Condenser	737,000	-	126,600	863,600
Distributed, Air-Cooled	690,700	-	169,800	860,500
Distributed, Water-Cooled, Evap	697,500	-	168,600	866,100
Secondary Loop, Evap Condenser	684,500	74,500	116,200	875,200
Advanced Self-Contained, Water-Cooled, Evap	867,700	-	180,600	1,048,300

Results for supermarket at Washington, DC location
*Heat rejection includes energy consumed by fans and pumps (for water loops and evaporative cooling)

Table 7-5. Impact of heat rejection on multiplex refrigeration

Location	Air-Cooled	Water-Cooled, Evaporative Tower	Water-Cooled, Dry Tower	Evaporative
Worcester, MA	904,500	916,100	1,025,800	836,100
Washington, DC	976,800	975,700	1,102,700	896,400
Memphis, TN	1,050,200	1,050,400	1,192,200	964,800
Los Angeles, CA	1,067,200	1,003,800	1,196,400	928,700

Table 7-6. Impact of heat rejection on distributed refrigeration

Location	Air-Cooled	Water-Cooled, Evaporative Tower	Water-Cooled, Dry Tower
Worcester, MA	785,700	802,200	898,600
Washington, DC	860,500	866,100	977,900
Memphis, TN	942,800	943,700	1,077,900
Los Angeles, CA	958,400	896,400	1,064,800

Table 7-7. Impact of heat rejection on secondary loop refrigeration

Location	Evaporative Condenser	Air-Cooled	Water-Cooled, Evaporative Tower	Water-Cooled, Dry Tower
Worcester, MA	821,600	880,500	900,000	1,266,700
Washington, DC	875,200	951,400	959,700	1,327,200
Memphis, TN	940,400	1,032,600	1,034,800	1,339,500
Los Angeles, CA	892,300	1,039,500	979,200	1,370,700

condensing were considered. Two types of heat rejection for the water-cooled condensing were evaluated; either evaporative or dry fluid coolers were used for final rejection. For the distributed refrigeration, air-cooled condensing, and water-cooled condensing using either evaporative or dry heat rejection were analyzed.

Table 7-8 shows the analysis results for low-charge multiplex refrigeration where air-cooled and evaporative condensing were examined. Energy savings are seen for low-charge multiplex for all locations. Savings are increased when evaporative condensing is employed. Evaporative condensing requires lower fan energy to maintain low condensing temperature operation. This difference in fan energy is a major portion of the savings increase. The remainder can be attributed to lower condensing temperature achieved through the use of evaporative condensing during warm weather operation.

Table 7-9 compares the energy consumption of multiplex with air-cooled condensing to distributed with dry heat rejection. The use of dry rejection increases the energy consumption of the distributed system significantly. The reason for this is the added temperature difference incurred in heat rejection, which raises the refrigeration condensing temperature. This increase has the largest impact during the summer months, raising the condensing temperature of the

Table 7-8. Estimated annual energy consumption for standard and low-charge multiplex refrigeration

Location	Multiplex System Type	Annual System Energy (kWh)	Low-Charge Savings	Percent Savings
Worcester, MA	Standard	904,600		
	Low-Charge Air-Cooled	850,000	54,600	6.0
	Low-Charge Evaporative Condenser	791,600	112,900	12.5
Washington, DC	Standard	976,800		
	Low-Charge Air-Cooled	935,200	41,400	4.3
	Low-Charge Evaporative Condenser	863,600	113,200	11.6
Memphis, TN	Standard	1,058,000		
	Low-Charge Air-Cooled	1,027,100	25,600	2.4
	Low-Charge Evaporative Condenser	941,500	116,400	11.0
Los Angeles, CA	Standard	1,067,200		
	Low-Charge Air-Cooled	1,042,600	24,600	2.3
	Low-Charge Evaporative Condenser	911,300	156,000	14.6

Table 7-9. Annual energy consumption comparison between multiplex with air-cooled condensing and distributed with dry heat rejection

	Refrigeration System		
Location	Multiplex (kWh)	Distributed (kWh)	Savings Achieved by Distributed (kWh)
Worcester, MA	904,600	898,600	6,000 (0.7%)
Washington, DC	976,800	977,900	-1,100
Memphis, TN	1,058,200	1,077,900	-19,700
Los Angeles, CA	1,067,200	1,064,800	2,400 (0.2%)

Multiplex employs air-cooled condensing.
Distributed employs water-cooled condensing with a dry fluid cooler.

distributed system above that of the multiplex. The higher condensing temperature also decreases the number of hours that the distributed system can operate at the minimum condensing temperature during winter months. Some energy savings were seen for the distributed system at two locations, Worcester, MA and Los Angeles, CA; but negative energy savings were predicted for operation in Washington, DC and Memphis, TN. These results show the value of evaporative heat rejection for close-loop cooling systems.

Table 7-10 compares energy consumption values of multiplex with evaporative condensing and distributed with evaporative heat rejection. An increased condensing temperature can still be expected for the distributed system because of the added temperature difference of the fluid loop. Some added energy consumption for the distributed system can be anticipated for summer operation because of increased condenser temperature. Winter operation is less likely to be impacted, and operation at minimum condensing temperature can be expected for most of the time. Energy savings range from 2.2 to 4.0 percent for this configuration. The largest savings are seen for operation in Worcester, MA, which has the coldest climate.

Table 7-11 compares the performance of multiplex and distributed refrigeration when both employ water-cooled condensing and evaporative heat rejection. In this situation, the temperature differences incurred in heat rejection are the same for both systems. Energy savings achieved by the distributed system ranged from 10.2 to 12.4 percent for the sites examined.

7.1.2 Analysis of Refrigeration Heat Reclaim for Space Heating

The initial analysis work done for refrigeration heat reclaim for space heating was to determine the correct value of minimum condensing temperature to be used to maximize savings obtained. Minimum condensing temperature value affects the amount of heat reclaimed, which increases as the condensing temperature increases. It also impacts the energy consumption of the refrigeration system, which will also increase as the condensing temperature is increased. In order to determine the best operating point, multiple analyses were conducted in which different values of minimum condensing temperature were employed. The annual energy consumption of the refrigeration system and the amount of heat reclaim were calculated at each of the

Table 7-10. Annual energy consumption comparison between multiplex with evaporative condensing and distributed with evaporative heat rejection

	Refrigeration System		
Location	Multiplex (kWh)	Distributed (kWh)	Savings Achieved by Distributed (kWh)
Worcester, MA	836,100	802,200	33,900 (4.0%)
Washington, DC	896,400	866,100	30,300 (3.4%)
Memphis, TN	964,800	943,700	21,100 (2.2%)
Los Angeles, CA	928,700	896,400	32,300 (3.5%)

Multiplex employs evaporative condensing.
Distributed employs water-cooled condensing with an evaporative fluid cooler.

Table 7-11. Annual energy consumption comparison between multiplex and distributed refrigeration with water-cooled condensing and evaporative heat rejection

	Annual Energy Consumption (kWh) Refrigeration System		
Location	Multiplex	Distributed	Savings Achieved by Distributed (kWh)
Worcester, MA	916,100	802,200	113,900 (12.4%)
Washington, DC	975,700	866,100	109,600 (11.2%)
Memphis, TN	1,050,400	943,700	106,700 (10.2%)
Los Angeles, CA	1,003,800	896,400	107,400 (10.7%)

Multiplex employs water-cooled condensing with an evaporative fluid cooler.
Distributed employs water-cooled condensing with an evaporative fluid cooler.

temperatures. A comparison of operating cost could then be made which includes the cost of electric energy to operate the refrigeration and the value of natural gas, which is displaced by the reclaimed heat for space heating.

Table 7-12 and Figure 7-2 give the results of such an analysis for a supermarket located in Worcester, MA. The results also include the no heat reclaim situation where the minimum condensing temperature is maintained at 70°F. The analysis showed that the amount of heat recovered is strongly influenced by the condensing temperature. The amount of heat reclaim seen at a 70°F condensing temperature is small and limited to desuperheating of the refrigerant only, due to the small temperature difference existing between the refrigerant and the circulated air. Heat recovery at 70°F is often at less than full desuperheating because of this small temperature difference. The combined operating costs for the heat reclaim and no heat reclaim systems are very close with a difference of only $783. At a minimum condensing temperature of 80°F, the amount of heat recovered is substantially more due to the increase in temperature difference at the heat reclaim coil. Full desuperheating is achieved at this condensing temperature. Savings achieved with heat reclaim at 80°F condensing are close to the largest savings seen. The graph in Figure 7-2 suggests that the optimum condensing temperature is approximately 80°F, where the lowest combined operating cost for refrigeration and heating is achieved. At higher condensing temperatures, more heat is reclaimed, but the amount of refrigeration energy consumed increases dramatically, so that no savings are seen at these higher condensing temperatures.

The same analysis was conducted for a supermarket operating in Los Angeles, CA. The results are shown in Table 7-13 and Figure 7-3. The results are similar to that seen for Worcester, where the lowest cost of operation is at a condensing temperature of 80°F. The savings achieved are considerably less than were seen at the Worcester site, making heat reclaim questionable for warm climates where space heating is of less significance.

Table 7-12. Heat reclaim performance multiplex refrigeration with air-cooled condensing Worcester, MA location

Minimum Condensing Temperature (ºF)	Annual Energy Consumption: Refrigeration (kWh)	Annual Energy Consumption: Heating Gas (MMBTU)	Annual Energy Consumption: Added Fan Energy (kWh)	Annual Operating Cost ($): Refrigeration	Annual Operating Cost ($): Heating	Annual Operating Cost ($): Total
70 (No Heat Reclaim)	904,600	3,155		83,223	23,158	106,381
70	904,600	2,675	29,800	83,223	22,376	105,599
80	954,600	1,940	29,800	87,823	16,981	104,804
90	1,006,700	1,803	29,800	92,616	15,976	108,592
100	1,090,700	1,626	29,800	100,344	14,676	115,021

Operating Costs based upon an electric cost of $0.092/kWh and a gas cost of $7.34/MMBTU.

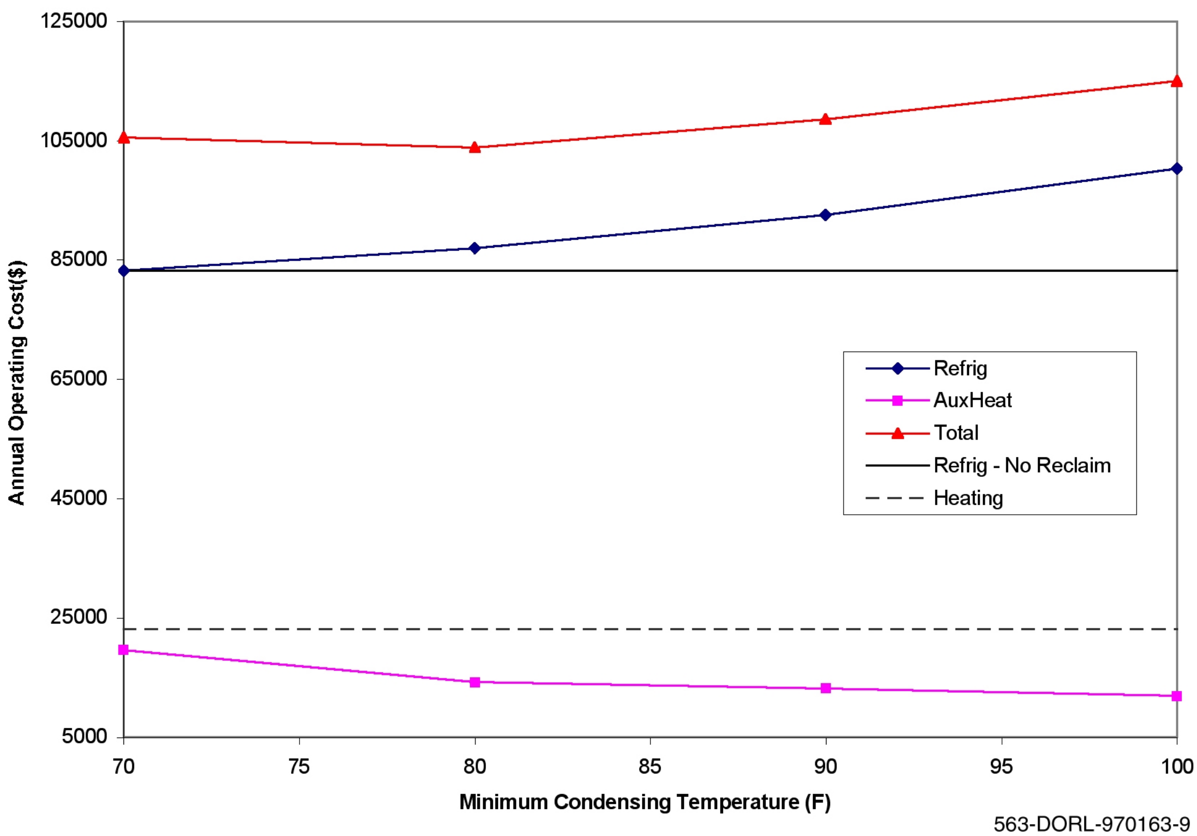

Figure 7-2. Analysis of heat reclaim for space heating, multiplex refrigeration with air-cooled condensing, Worcester, MA location

Table 7-13. Heat reclaim performance multiplex refrigeration with air-cooled condensing Los Angeles, CA location

Minimum Condensing Temperature (ºF)	Annual Energy Consumption: Refrigeration (kWh)	Heating Gas (MMBTU)	Added Fan Energy (kWh)	Annual Operating Cost ($): Refrigeration	Heating	Total
70 (No Heat Reclaim)	1,067,200	882		98,182	5,671	103,854
70	1,067,200	610	29,784	98,182	6,662	104,845
80	1,098,400	79	29,784	101,053	3,248	104,301
90	1,141,300	55	29,784	105,000	3,094	108,093
100	1,217,600	29	29,784	112,019	2,927	114,946

Operating Costs based upon an electric cost of $0.092/kWh and a gas cost of $6.43/MMBTU.

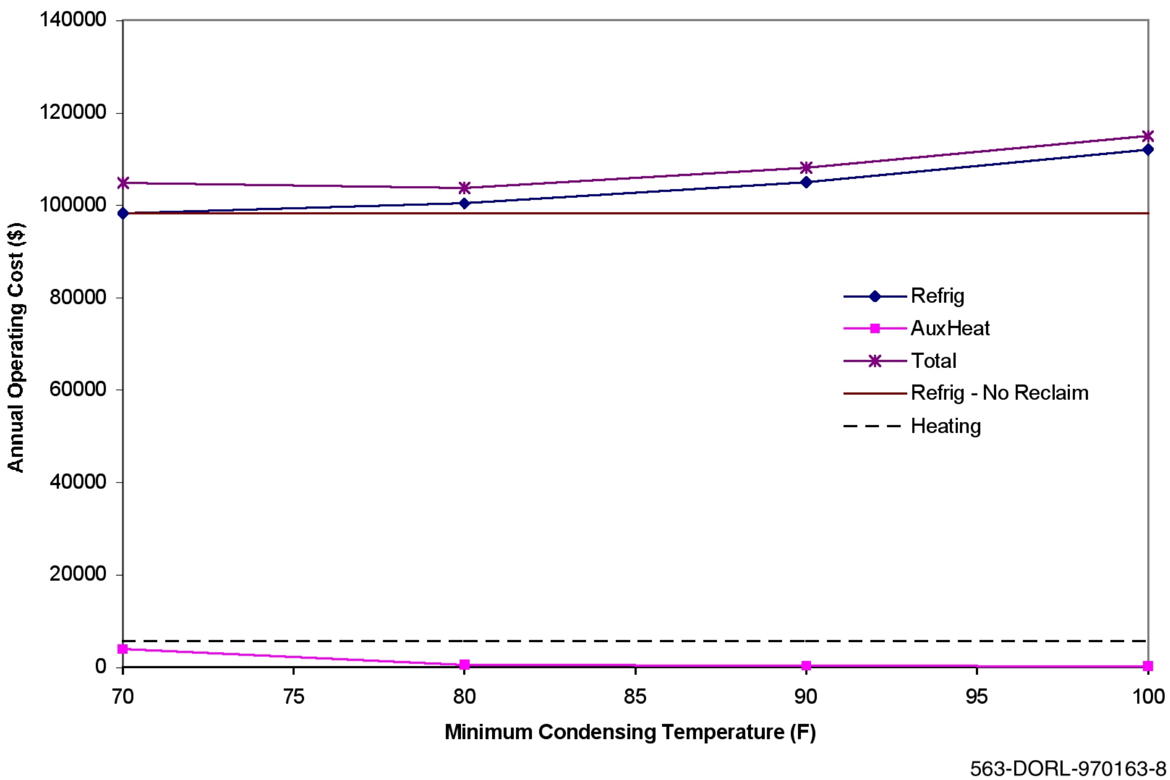

Figure 7-3. Analysis of heat reclaim for space heating, multiplex refrigeration with air-cooled condensing, Los Angeles, CA location

7.2 Evaluation of Environmental Impact of Low Charge Refrigeration

The environmental benefit that can be derived by the use of advanced, low charge refrigeration is a significant reduction in the amount of halogenated refrigerants now used in supermarkets. Present supermarkets employ as much as 3,000 lb of refrigerant, most of which is HCFC-22. This refrigerant has an ozone depletion potential (ODP) of 0.055 and a global warming potential (GWP) of 1700. The latest replacement refrigerants are HFCs, such as R-134a, R-404A, and R-507, which have an ODP of 0, but have high GWP values, in the range of 1300, 3260, and 3300, respectively (7-1).

All refrigeration systems considered here offer better approaches in terms of reduction and containment of refrigerant. Some variation in charge requirement is seen depending upon the type of heat rejection employed. Lowest charge is required by systems employing a fluid loop for heat rejection. The charge requirement for close-coupled systems, such as the distributed, secondary loop, and advanced self-contained is less due to the reduction in suction-side piping. The estimated charge for a distributed refrigeration system with fluid-loop heat rejection is approximately 900 lb, which is based on a charge requirement for each compressor cabinet of 90 lb, and, typically, 10 cabinets are needed in a supermarket. The charge associated with the secondary loop refrigeration system employing evaporative condensing is approximately 500 lb, which is split between two chiller systems. The charge requirement of the secondary loop sysstem can be reduced further to approximately 200 lb if water-cooled condensers and a fluid loop are used for heat rejection. For either system, the significant reduction in refrigerant piping drastically reduces the annual leakage rate to no more than 5 percent of total charge annually.

The environmental impact of the supermarket refrigeration system, including both the refrigerant charge and energy consumption can be determined through the use of the total equivalent warming impact (TEWI), which is a measure of the direct impact of refrigerant emissions and the indirect impact of electric generation on global warming. The direct portion of the TEWI shows the effect of refrigerant emissions on global warming due to the atmospheric lifetime of the refrigerant. The indirect portion shows the emission of carbon dioxide during generation of electric energy to drive the refrigeration system.

The TEWI can be calculated on an annual basis from the relation:

$$TEWI = Mass_{ref} * GWP_{ref} + E_{annual} * C$$

where

$Mass_{ref}$	=	the amount of refrigerant leaking from the system annually (kg)
GWP_{ref}	=	Global Warming Potential of the refrigerant
E_{annual}	=	the annual electric energy consumption of the refrigeration system
C	=	the emission rate of CO_2 associated with electric generation. For North America, the accepted value is 0.65 kg of CO_2 per kWh

Table 7-14 shows the results of the TEWI calculations for the refrigeration systems considered here. The estimates are given for a system life of 15 years at a location in Washington, DC. The system leak rates were taken from TEWI investigation conducted by Oak Ridge National Laboratory (7-2).

The results show that the TEWI values for the distributed and secondary loop systems are significantly lower than that of the multiplex system. The lowest TEWI value is achieved by the distributed system employing a fluid loop for heat rejection. The low-charge multiplex system shows some reduction in TEWI versus the baseline multiplex, due to the reductions in system charge and energy use. The advanced self-contained systems has the lowest direct TEWI value, but has the largest indirect value due to high energy consumption.

Table 7-15 gives the estimated operating savings for the low-charge systems due to reduced refrigerant leakage. For this analysis, refrigerant costs of $1.75/lb for R-22, and $7.75/lb for R-404A and R-507, were used, respectively.

7.3 Analysis Results for HVAC

The results for the analysis of supermarket HVAC systems are shown in Tables 7-16 and 7-17. Table 7-16 shows the annual electric and gas consumption for a conventional HVAC

Table 7-14. Total Equivalent Warming Impact (TEWI) for supermarket refrigeration

						TEWI (million kg of CO_2)		
System	Condensing	Charge (lb)	Refrigerant	Leak (%)	Annual Energy (kWh)	Direct	Indirect	Total
Multiplex	Air-Cooled	3,000	R404A/	30	976,800	13.62	9.52	23.15
	Evaporative	3,000	R-22	30	896,400	13.62	8.74	22.36
Low-Charge	Air-Cooled	2,000	R404A/	15	935,200	4.54	9.12	13.66
Multiplex	Evaporative	2,000	R-22	15	863,600	4.54	8.42	12.96
Distributed	Air-Cooled	1,500	R404A	10	860,500	3.33	8.38	11.71
Distributed	Water-Cooled, Evaporative	900	R404A	5	866,100	1.00	8.44	9.44
Secondary Loop	Evaporative	500	R507	10	875,200	1.13	8.54	9.67
Secondary Loop	Water-Cooled, Evaporative	200	R507	5	959,700	0.23	9.36	9.58
Advanced Self-Contained	Water-Cooled, Evaporative	100	R404A	1	1,048,300	0.02	10.22	10.24

Results for site in Washington, DC – 15 year service life.
Conversion factor = 0.65 kg CO_2/kWh.
Multiplex – 33.3% R404A (low temperature), GWP = 3260; 66.7% R22 (medium temperature), GWP = 1700.
Distributed and Advanced Self-Contained – 100% R404A, GWP = 3260.
Secondary Loop – 100% R507, GWP = 3300.

Table 7-15. Estimated operating cost savings for reduced refrigerant leakage

System	Annual Leakage (lb) R-404A	R-507	R-22	Savings ($)
Multiplex - (R-404A/R-22)	300		600	
Multiplex - Low Charge	100		200	2,250
Multiplex - Low Charge Evap Cond	100		200	2,250
Distributed Air-Cooled	150			2,213
Distributed Water-Cooled, Evap	45			3,026
Secondary Loop Evap Condensing		50		2,988
Secondary Loop Water-cooled, Evap		10		3,298
Advanced Self-Contained	1			3,367

Table 7-16. Annual energy consumption for conventional supermarket HVAC

Location	System	Electric Consumption (kWh) Reclaim	HVAC Fans	Cooling	Total	Gas Consumption (MMBTU)
Worcester, MA	Conventional	0	160,307	45,498	205,805	3,155
	Heat Reclaim	40,200	190,100	45,498	275,798	1,894
Washington, DC	Conventional	0	160,307	68,085	228,392	2,392
	Heat Reclaim	30,200	190,100	68,085	288,385	1,215
Memphis, TN	Conventional	0	160,307	119,458	279,765	1,636
	Heat Reclaim	31,600	190,100	119,458	341,158	637
Los Angeles, CA	Conventional	0	160,307	43,207	203,514	882
	Heat Reclaim	24,000	190,100	43,207	257,307	72

Table 7-17. Annual energy consumption for water-source heat pump HVAC

Location	Electric Energy Consumption (kWh) HVAC Fans	Water-source Heat Pump Cooling	Heating	Loop Pump	Total	Gas Consumption (MMBTU)
Worcester, MA	160,307	35,920	120,004	26,280	342,511	12
Washington, DC	160,307	54,429	91,095	26,280	332,112	0
Memphis, TN	160,307	96,656	60,998	26,280	344,241	0
Los Angeles, CA	160,307	35,321	32,920	26,280	254,829	0

system, consisting of rooftop units alone, and for rooftop units with refrigeration heat reclaim for space heating. The electric consumption is divided into the following categories:

- Reclaim – added energy used by the refrigeration system for heat reclaim.

- HVAC fans – energy consumed for air circulation through the store, including fan energy associated with the heat reclaim heating coil.

- Cooling – energy consumed by the condensing units used to provide air conditioning.

- Gas consumption - consists of gas used by duct heaters to provide space heating.

Table 7-17 gives the annual energy consumption for HVAC provided by water-source heat pumps. Energy consumption consists of fan energy for air circulation (same as conventional system), operation of the heat pump compressors, and for water pumping. Fan energy for heat rejection is accounted for in the consumption of the refrigeration system, since the same water loop is used for both systems. Gas consumption represents heating needed to supplement the output of the water-source heat pumps.

Table 7-18 gives the operating cost associated with the energy use described above. Local utility rates were used for each location to estimate the cost. Water cost for evaporative heat rejection was included in the cooling costs associated with the water-source heat pumps. A breakdown of the operating cost is also provided. The largest element of the HVAC operating cost is the cost of energy for the circulation fans which accounts for 35 to 67.9 percent of the total cost. Heating also represents are large fraction of the cost in two locations, Worcester and Washington. In Memphis and Los Angeles, the heating and cooling costs are similar in magnitude.

Figure 7-4 shows these cost breakdowns graphically. The results indicate that the water-source heat pump system operates with the lowest annual cost in all locations considered. Table 7-19 gives the savings obtained by the water-source heat pumps when compared to conventional HVAC and conventional HVAC with heat reclaim. Savings range from 2.2 to 30.7 percent versus conventional HVAC alone, and from 3.8 to 22.1 percent versus conventional HVAC with heat reclaim for space heating. The largest savings were seen for a Washington, DC location, which had a combination of large heating load and favorable utility rates. The smallest savings versus heat reclaim were seen for a Memphis, TN location, which was due to unfavorable utility rates, primarily a low rate for natural gas. HVAC savings for a Los Angeles location were small for either heat reclaim or water-source heat pumps, because the space heating requirement for this location is smallest of all sites considered.

7.4 Analysis of Integrated Operation of Refrigeration and HVAC

An analysis was performed to determine the best integrated system approach to refrigeration and HVAC. The systems that were compared were:

Table 7-18. Annual operating costs for supermarket HVAC

Location	System	Reclaim $	Reclaim %	Fans $	Fans %	Cooling $	Cooling %	Heating $	Heating %	Total $
Worcester, MA	Conventional	0	0.0	14,748	35.0	4,186	9.9	23,158	55.0	42,092
	Heat Reclaim	4,609	11.5	17,488	43.5	4,186	10.4	13,902	34.6	40,185
	WSHP	0	0.0	14,748	46.7	5,975	14.3	12,337	39.0	33,060
Washington, DC	Conventional	0	0.0	11,382	33.6	4,834	14.3	17,653	52.1	33,869
	Heat Reclaim	2,833	9.4	13,496	44.8	4,834	16.0	8,967	29.8	30,130
	WSHP	0	0.0	11,382	50.2	5,605	21.2	6,490	28.6	23,477
Memphis, TN	Conventional	0	0.0	10,260	36.8	7,645	27.4	9,996	35.8	27,901
	Heat Reclaim	2,534	9.4	12,166	45.3	7,645	28.5	4,503	16.8	26,849
	WSHP	0	0.0	10,260	42.2	8,293	28.9	7,027	28.9	25,580
Los Angeles, CA	Conventional	0	0.0	14,748	60.5	3,975	16.3	5,671	23.2	24,395
	Heat Reclaim	2,870	11.6	17,488	70.5	3,975	16.0	463	1.9	24,797
	WSHP	0	0.0	14,748	65.7	4,641	14.5	4,458	19.9	23,848

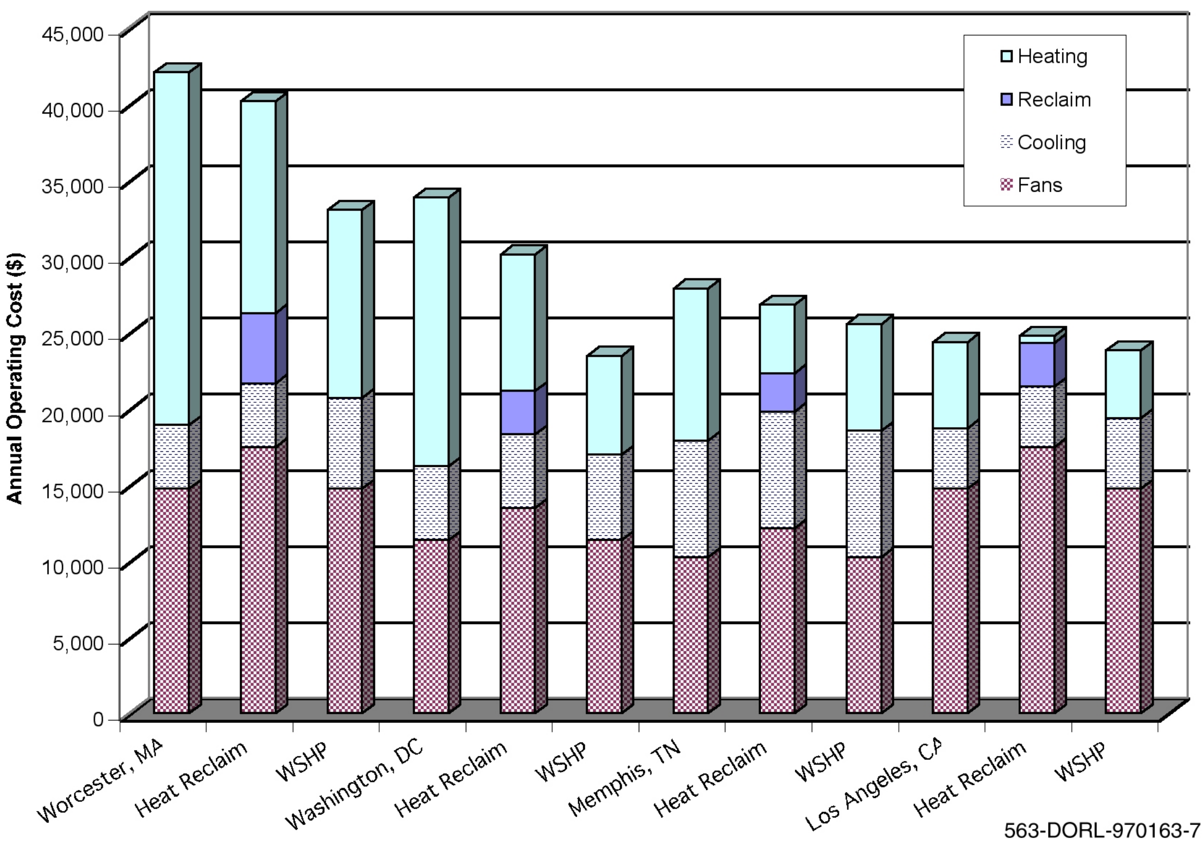

Figure 7-4. Comparison of supermarket HVAC systems

Table 7-19. HVAC operating cost savings achieved by water-source heat pumps

Location	Annual Savings ($) versus Conventional	vesus Heat Reclaim
Worcester, MA	9,032 (21.4%)	7,125 (17.7%)
Washington, DC	10,391 (30.7%)	6,653 (22.1%)
Memphis, TN	2,321 (8.3%)	1,269 (4.7%)
Los Angeles, CA	547 (2.2%)	949 (3.8%)

- Multiplex refrigeration with conventional rooftop units for HVAC.
- Multiplex refrigeration with heat reclaim and conventional HVAC.
- Distributed refrigeration with water-source heat pumps for HVAC.

Table 7-20 and Figure 7-5 show the estimated operating costs for refrigeration and HVAC for the sites examined. The lowest cost approach was the combination of distributed refrigeration and water-source heat pumps for all four locations. Table 7-21 gives the annual savings achieved for each site. The savings ranged from a low of 11.1 percent for Memphis, TN to a high of 19.2 percent for Washington, DC. Table 7-21 also contains a comparison of multiplex refrigeration and conventional HVAC with and without heat reclaim. Savings were seen for heat reclaim operation at all locations except Los Angeles. Savings achieved by heat

Table 7-20. Estimated annual operating cost for supermarket refrigeration and HVAC

		Annual Operating Cost ($)				
		Energy				
Location	System	Refrigerant	HVAC	Refrigerant	Water	Total
Worcester, MA	Multiplex	83,214	42,092	3,375	0	128,681
	Multiplex and Heat Reclaim	86,912	35,576	3,375	0	125,864
	Distributed and WSHP	73,802	31,599	349	2,922	108,672
Washington, DC	Multiplex	69,353	33,869	3,375	0	106,597
	Multiplex and Heat Reclaim	72,193	27,297	3,375	0	102,865
	Distributed and WSHP	61,493	22,669	349	1,616	86,127
Memphis, TN	Multiplex	67,213	27,901	3,375	0	98,489
	Multiplex and Heat Reclaim	69,231	24,314	3,375	0	96,921
	Distributed and WSHP	60,365	24,314	349	2,533	87,560
Los Angeles, CA	Multiplex	98,182	24,395	3,375	0	125,952
	Multiplex and Heat Reclaim	101,047	21,926	3,375	0	126,348
	Distributed and WSHP	82,285	22,456	349	2,783	107,873

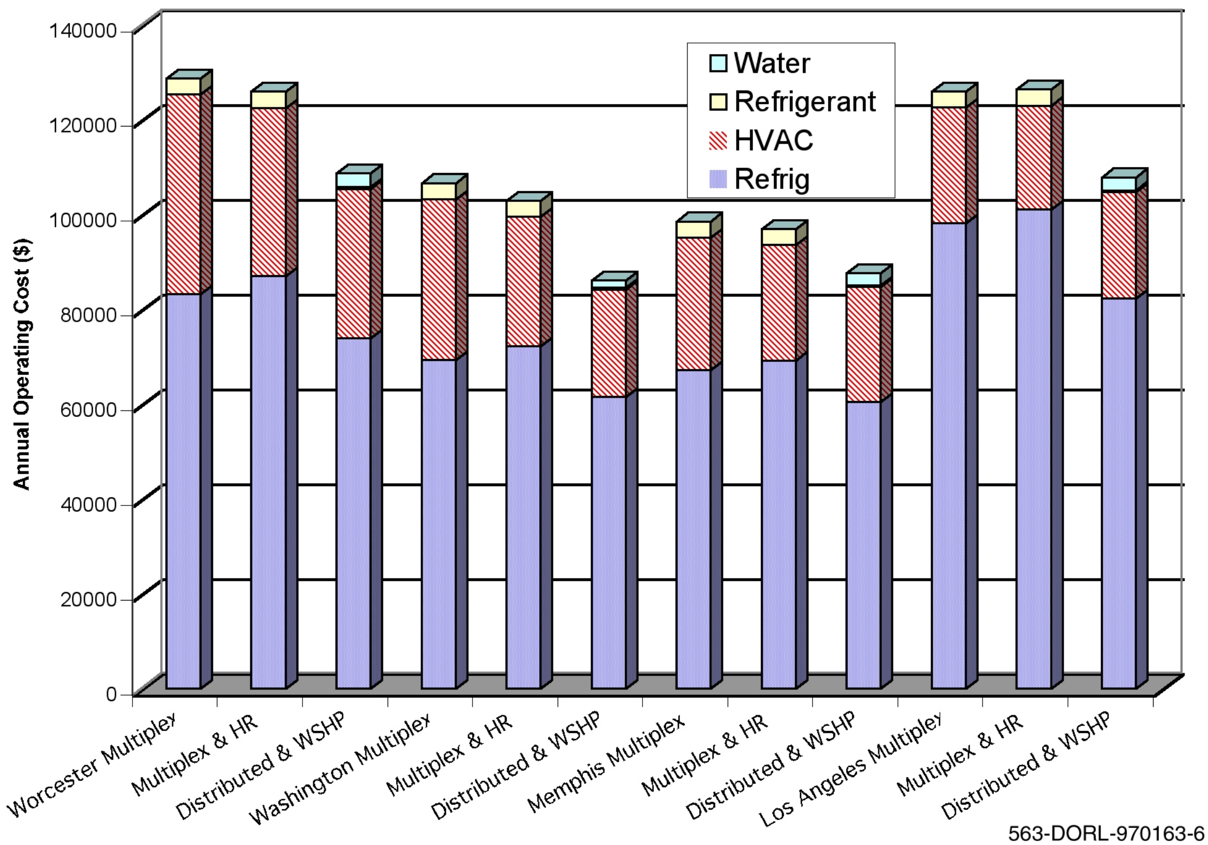

Figure 7-5. Operating cost of supermarket refrigeration and HVAC

Table 7-21. Annual cost savings achieved by distributed refrigeration and water-source heat pumps versus multiplex refrigeration and conventional HVAC

	Annual Operating Savings			
	with Heat Reclaim		Distributed Refrigeration and WS Heat Pumps	
Location	$	%	$	%
Worcester, MA	2,817	2.2	20,009	15.5
Washington, DC	3,732	3.5	20,469	19.2
Memphis, TN	1,568	1.6	10,929	11.1
Los Angeles, CA	-397	-0.3	18,079	14.4

reclaim were considerably less than seen with distributed refrigeration and water-source heat pumps, ranging from -0.3 to 3.5 percent.

7.5 Payback Analysis for Advanced Systems

Table 7-22 gives the estimated installed cost premiums for distributed, secondary loop, and low-charge multiplex refrigeration systems. Estimates are based on actual construction budgets

Table 7-22. Estimated installed cost premiums for low-charge supermarket refrigeration systems

System	Installed Cost Premium ($) Equipment	Installation	Total
Multiplex	Baseline		
Low-charge Multiplex	0	0	0
Distributed, Water-cooled	53,000	7,000	60,000
Secondary Loop, Evaporative	70,000	77,000	147,000

supplied by the engineering departments of two supermarket chains. For purposes of confidentiality between the chains and their vendors, the identities of these supermarkets will not be given in this report. It should be noted that the actual installed cost of any refrigeration system will vary greatly, depending upon many factors, such as: purchasing arrangements between the supermarket and refrigeration equipment vendors; whether or not display cases are purchased in conjunction with the refrigeration system; special system features or configuration requested by the supermarket; and unique installation requirements of each site. The lowest cost premium is seen with the low-charge multiplex system, which was estimated to have the same installed cost as the baseline multiplex system. The distributed system showed higher equipment cost, but only a small increase in installation cost. This can be attributed to reduced refrigeration piping cost, but also increased electrical and fluid loop costs. The equipment and installation cost premiums of the secondary loop system are similar in magnitude.

The cost premium for the water-source heat pumps is shown in Table 7-23 and is estimated at $25,000, which consists of $15,000 in added equipment cost and $10,000 in installation cost. The extra installation cost includes water piping for the heat pumps and over-sizing of the refrigeration heat rejection to allow heat pump heat rejection during space cooling.

Table 7-24 gives the estimated simple paybacks for each of the low-charge systems at each of the sites examined. Operating cost savings for the refrigeration include the savings obtained for reduced energy use and refrigerant leakage. Water costs are included for all systems employing evaporative heat rejection. The low-charge multiplex achieves immediate payback, since no installed cost premium exists. Highest savings for the low-charge multiplex are seen when evaporative condensing is employed. Paybacks for the distributed system ranged from 3.4 to 7.0 years, while the payback for the secondary loop system ranged from 8.3 to 16.8 years.

Table 7-25 shows the payback associated with the use of distributed refrigeration in combination with water-source heat pumps for store HVAC. The combined payback for refrigeration and

Table 7-23. Estimated installed cost premiums for water-source heat pumps

HVAC (Water-source heat pumps)	Cost ($)
Equipment	15,000
Installation	10,000
Total Installed Cost Premium for HVAC	25,000

Table 7-24. Estimated energy savings and payback for low-charge supermarket refrigeration (versus multiplex with air-cooled condensing)

Location	Low-Charge Multiplex, Air-Cooled $	Low-Charge Multiplex, Air-Cooled Year	Low-Charge Multiplex, Evaporative $	Low-Charge Multiplex, Evaporative Year	Distributed, Water-Cooled, Evaporative $	Distributed, Water-Cooled, Evaporative Year	Secondary Loop, Evaporative $	Secondary Loop, Evaporative Year
Worcester, MA	7,264	0	11,176	0	10,977	5.5	9,153	16.1
Washington, DC	5,204	0	9,479	0	10,078	6.0	9,393	15.6
Memphis, TN	3,728	0	7,940	0	8,608	7.0	8,748	16.8
Los Angeles, CA	4,513	0	15,201	0	17,532	3.4	17,678	8.3

Table 7-25. Estimated payback for distributed refrigeration and water-source heat pumps

Location	Savings ($) Refrigeration	Savings ($) Combined	Payback (Year) Refrigeration	Payback (Year) Combined
Worcester, MA	10,977	20,009	5.5	4.2
Washington, DC	10,078	20,469	6.0	4.2
Memphis, TN	8,608	10,929	7.0	7.8
Los Angeles, CA	17,532	18,079	3.4	4.7

HVAC savings was less than that for refrigeration savings alone for the Worcester and Washington sites, because of increased space heating savings. Combined paybacks for the Memphis and Los Angeles sites were longer, because space heating is not as significant at these locations.

Section 7 References

7-1. IPCC 1995. Climate Changes 1995: The Science of Climate Change*, Working Group I, Cambridge University Press, 1996.*

7-2. *Sand, James R, Steven K. Fischer, Van D. Baxter,* Energy and Global Warming Impacts of HFC Refrigerants and Emerging Technologies*, Oak Ridge National Laboratory, sponsored by Alternative Fluorocarbons Environmental Acceptability Study (AFEAS), U.S. Department of Energy, 1997.*

8. CONCLUSIONS AND RECOMMENDATIONS

The investigation of low charge supermarket refrigeration showed that at present, four system approaches are available, consisting of:

- Distributed refrigeration – compressors cabinets located throughout the store and a fluid loop for heat rejection.
- Secondary loop refrigeration – central chiller is used to cool a secondary fluid that is pumped to the display cases to provide refrigeration.
- Advance self-contained - each display case is equipped with a water-cooled condensing unit. A fluid loop is connected at all condensing units and is used for heat rejection.
- Low-charge multiplex - the multiplex refrigeration system is equipped with control piping and valves to allow operation at close to critical charge, greatly reducing the amount of refrigerant needed.

All of these advanced systems provide substantial reductions in refrigerant charge, and energy savings versus multiplex systems were shown for all systems with the exception of the advanced self-contained.

Further investigation and analysis was performed to predict the energy consumption for these low-charge systems and compare their performances to multiplex refrigeration with air-cooled condensing and mechanical subcooling for low temperature refrigeration, which is now the most commonly installed supermarket refrigeration system. Results from this analysis showed that the largest energy savings were achieved by the distributed and secondary loop refrigeration systems. The distributed system produced the energy savings, ranging from 10.2 to 16.2 percent of multiplex consumption. Secondary loop refrigeration reductions in energy of 9.2 to 16.4 percent for the locations investigated. The secondary loop system had higher energy savings than the distributed system for the Memphis and Los Angeles sites, while the distributed showed lower energy consumption for the Worcester and Washington sites. The low-charge multiplex system showed less energy use than the multiplex baseline for all locations. Savings ranged from 2.2 to 6.0 percent and 10.4 to 14.6 percent for the low-charge multiplex with air-cooled and evaporative condensing, respectively.

The energy savings achieved by the distributed refrigeration system can be attributed to close-coupling of the compressors to the display case evaporators, operation of the scroll compressors at 60°F minimum condensing temperature, and the use of evaporative heat rejection with the fluid loop. Savings seen with the secondary loop system are due to close-coupling of the compressors and the chiller evaporator, subcooling produced by brine heating for defrost, and the use of evaporative condensing. The refrigeration energy of the secondary loop system was found to be less than that of the distributed system, but the added energy associated with brine pumping negated some of this advantage. The energy savings seen with the low-charge multiplex system were due to the ability of this system to operate at very low minimum condensing temperatures. The minimum condensing temperatures were 40 and 60°F for low and medium temperature refrigeration, respectively.

A TEWI analysis of the low-charge refrigeration systems showed that both the distributed and secondary loop systems produced TEWI values that were significantly smaller than that estimated for the multiplex. Lowest TEWI values were achieved when a fluid loop with evaporative heat rejection was employed.

Substantial operating savings were obtained by the low-charge systems through reduced refrigerant leakage. The largest refrigerant savings were seen with the advanced self-contained system which was credited with a savings of \$3,367. Distributed and secondary loop systems produced refrigerant savings of \$3,026 and \$2,988, respectively. The smallest refrigerant savings were seen with the low-charge multiplex system at \$2,250.

A payback analysis of operating costs (electric, refrigerant, and water) for the low-charge refrigeration systems showed that the low-charge multiplex system had an immediate payback, since no installed cost difference exists between the low-charge and baseline multiplex systems. The distributed system showed paybacks ranging from 3.4 to 7.0 years, while the secondary loop system showed paybacks of 8.3 to 16.8 years. These payback values are extremely sensitive to the installed cost premium for these systems, which is highly variable depending upon arrangements between the supermarkets and their equipment suppliers and installers. These cost differences are likely to be reduced for either the distributed or the secondary loop systems as more such systems are implemented.

Supermarket HVAC systems were also addressed in this report where conventional rooftop HVAC, refrigeration heat reclaim, and water-source heat pumps were considered. The lowest operating cost for HVAC was shown for water-source heat pumps, which produced cost savings of 8.3 to 30.7 percent for the four locations examined.

For combined operation of refrigeration and HVAC, the system consisting of distributed refrigeration and water-source heat pumps showed the lowest operating cost for all locations considered. Operating cost savings were estimated to be 11.1 to 19.2 percent when compared to multiplex refrigeration with conventional HVAC. The payback on cost premium for distributed refrigeration and water-source heat pumps was found to be about 4.2 years for Worcester, MA and Washington, DC, and 4.7 years for Los Angeles, CA. The simple payback operation in Memphis, TN was 10.8 years. The lowest paybacks were seen for sites with large space heating loads. For these locations the operation of the water-source heat pumps helped to reduce the combined payback of the refrigeration and HVAC systems.

The results seen in this investigation show that low-charge refrigeration systems can reduce energy and operating costs if properly designed and operated. Demonstration of these technologies by field testing, possibly in conjunction with water-source heat pumps for HVAC, will help develop best practices for these systems and also better quantify energy savings. This information will help to accelerate the use of low-charge refrigeration systems by the supermarket industry.